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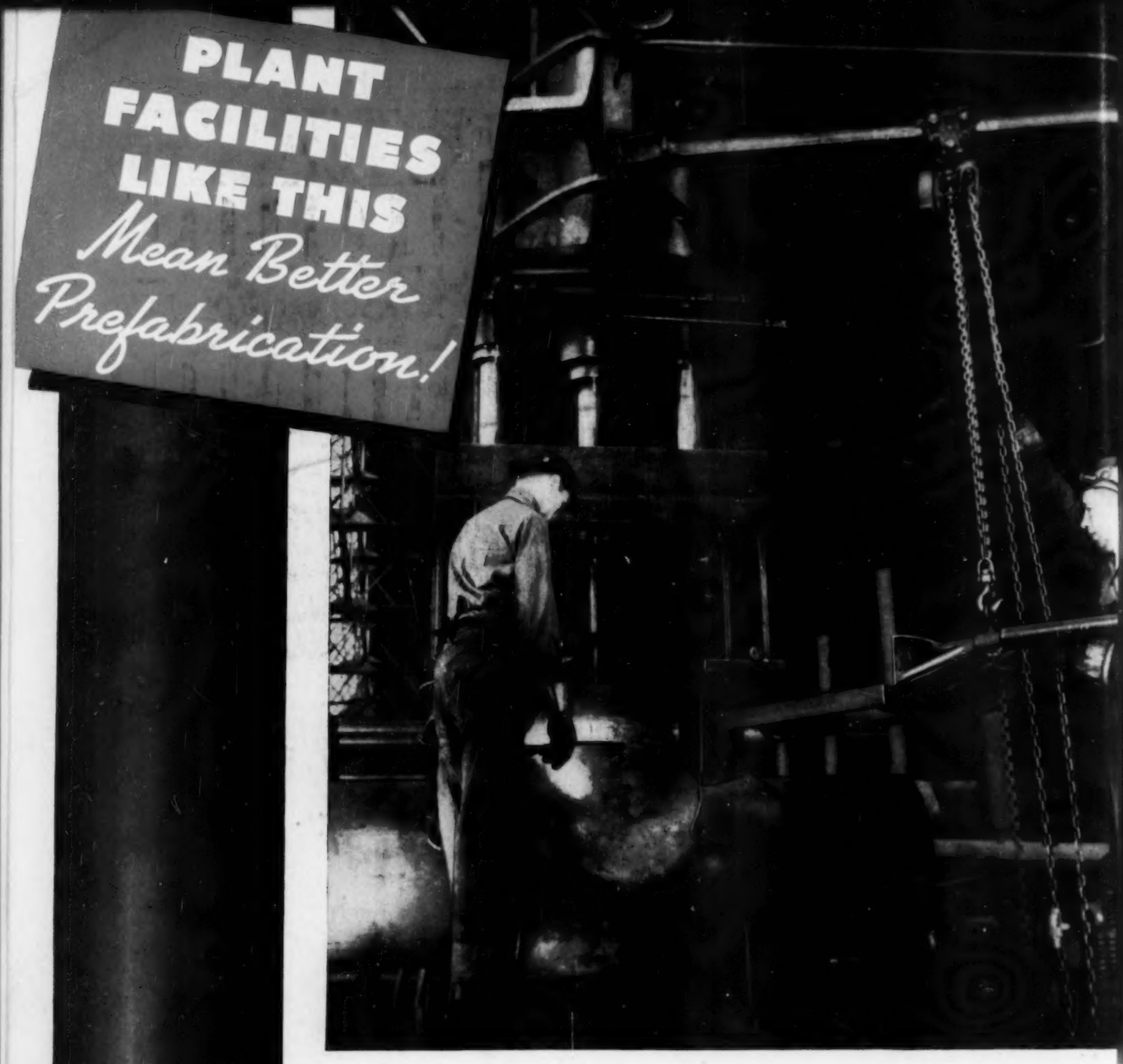

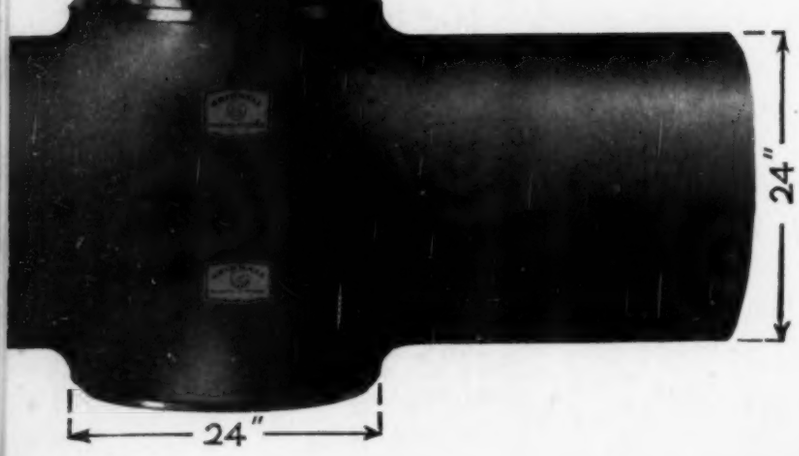

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The Editor's Page

■ **Each January,** HPAC features its Directory Section of Heating, Piping and Air Conditioning Equipment for Industry and Large Buildings. This year—as before—the Directory Section is in three parts—products classified, trade names, and manufacturers' addresses. The Directory Section which appears in this issue has been completely revised and brought up to date, which involved each company appearing therein checking its own listings and notifying our staff of any new products added or old ones dropped since last January. It is believed our readers will find the January, 1939, Directory Section of even greater usefulness than the earlier sections. . . . . In addition to the Directory Section, you will find in this month's HPAC the regular quota of editorial articles, as well as the Journal Section of the ASHVE. . . . . Another feature each January (and appearing in this issue) is our review of number and horsepower of air conditioning installations during the past year, as reported by the electric utilities in representative cities. . . . . This tabulation is supplemented this year by an article by Knight Porter and William Rock in which is analyzed 1938 air conditioning volume by types of installations in a large city.

■ **Construction contracts** to be let in the 37 Eastern states during 1939 may reach a total of $3,500,000,000, according to estimates released by F. W. Dodge Corp. This figure would compare with an estimated total of $3,200,000,000 for 1938, and an actual total of approximately $2,900,000,000 in 1937. None of these figures include repair and maintenance work. . . . . This national construction news organization points out that the total building and engineering volume recorded from January 1 through November 15 of 1938 reached a total of $2,635,000,000 (compared with $2,599,000,000 in the corresponding period of 1937). For 1939, indications of increased volume in public and private residential building, commercial and factory building, and public utility construction are stated to be very good. . . . . Reviewing such factors as the recent marked upturn after the 1937 recession, increased employment, the revival of business optimism after a period of excessive fears, and the widely held view that recent election results are favorable to business progress, Thomas S. Holden, vice-president in charge of statistics and research in the Dodge organization, states: "All these favorable factors might conceivably lead to a sustained construction revival to continue without serious interruption for several years. Yet, such a course is by no means certain, and present optimism should be tempered with reasonable caution." . . . . According to Holden's view, three potential factors would tend to curb recovery, if they should become actual in 1939. First, there is the question whether private building and engineering work will increase fast enough to offset the tapering off of PWA projects now scheduled for the second half of the year. Second, concurrent rises next spring in public and private housing activity, public works projects, and new industrial developments stimulated by a defense program might bring about increases in construction costs so rapidly as to curtail private operations. The third factor is the possibility of international difficulties; a new war threat would upset all present calculations and, lacking that, international trade complications might react unfavorably on American business and thus indirectly upon construction. . . . . The Dodge estimates for 1939 indicate that residential building, commercial building, and public utility construction may each increase about one-third over 1938, and that factory construction might double 1938's very low volume. Public and institutional buildings and public engineering works would decline appreciably unless there should be a new federal public works program.

■ **The twenty-fifth** anniversary of John Howatt's appointment as chief engineer of Chicago's Board of Education was celebrated by his friends and business associates at a testimonial dinner in the grand ball room of the Stevens hotel, Chicago, the evening of December 17. Some 1500 people turned out to honor the "Chief", well known to the heating and ventilating profession as past president of the ASHVE, and stayed to enjoy with him and Mrs. Howatt the dancing and entertainment. A complete movie camera and projector outfit was given to Mr. Howatt, the presentation being made by Mayor Edward J. Kelly. The affair was held under the auspices of the School Engineers' Union, Local No. 143, International Union of Operating Engineers. . . . Mr. Howatt entered the service of the Board of Education as chief engineer by virtue of a competitive civil service examination on December 16, 1913. Although a young man, he had already achieved a notable record as a student and engineer. After graduating with high honors from the engineering course at Minnesota, his experience included superintending the electrical installation of the U. S. Battleship Nebraska at the Seattle navy yard, two years as a consulting engineer, and several years as supervising engineer of all federal buildings from the Alleghenies to the Rockies and from Canada to the Gulf. . . . . Mr. Howatt is a member of HPAC's board of consulting and contributing editors.

■ **There is a great** need for a critical survey of design and operations of existing exhaust systems in industry, according to Prof. Philip Drinker of Harvard, chairman of the Air Hygiene Foundation's preventive engineering committee (and member of HPAC's board of consulting and contributing editors), in his report at the Foundation's annual meeting recently. "Such a study," he said, "would bring out fundamental requirements of design, particularly with respect to air velocities through hoods, transporting velocities, etc., as past experience needs to be studied critically in order to provide a basis for future design." The committee recommended that a full time fellowship be established to make such a study as it was felt the results would be of direct benefit in practical plant design, as well as in the development of codes and recommended standards of design. . . . . More than 200 executives from 101 industrial concerns attended the meeting, which was held at Mellon Institute in Pittsburgh. They heard reports on the latest advances in protecting the health of industrial workmen, who lose an estimated 200 million working days yearly because of illness.

■ **A conference** on air conditioning is to be held at the University of Illinois, Urbana, on March 8 and 9, under the sponsorship of the department of mechanical engineering and the engineering experiment station. . . . . Purpose of the conference is to present technical information on current problems and apparatus involved in small and medium size all year comfort air conditioning installations. . . . . Prof. W. H. Severns is the chairman.

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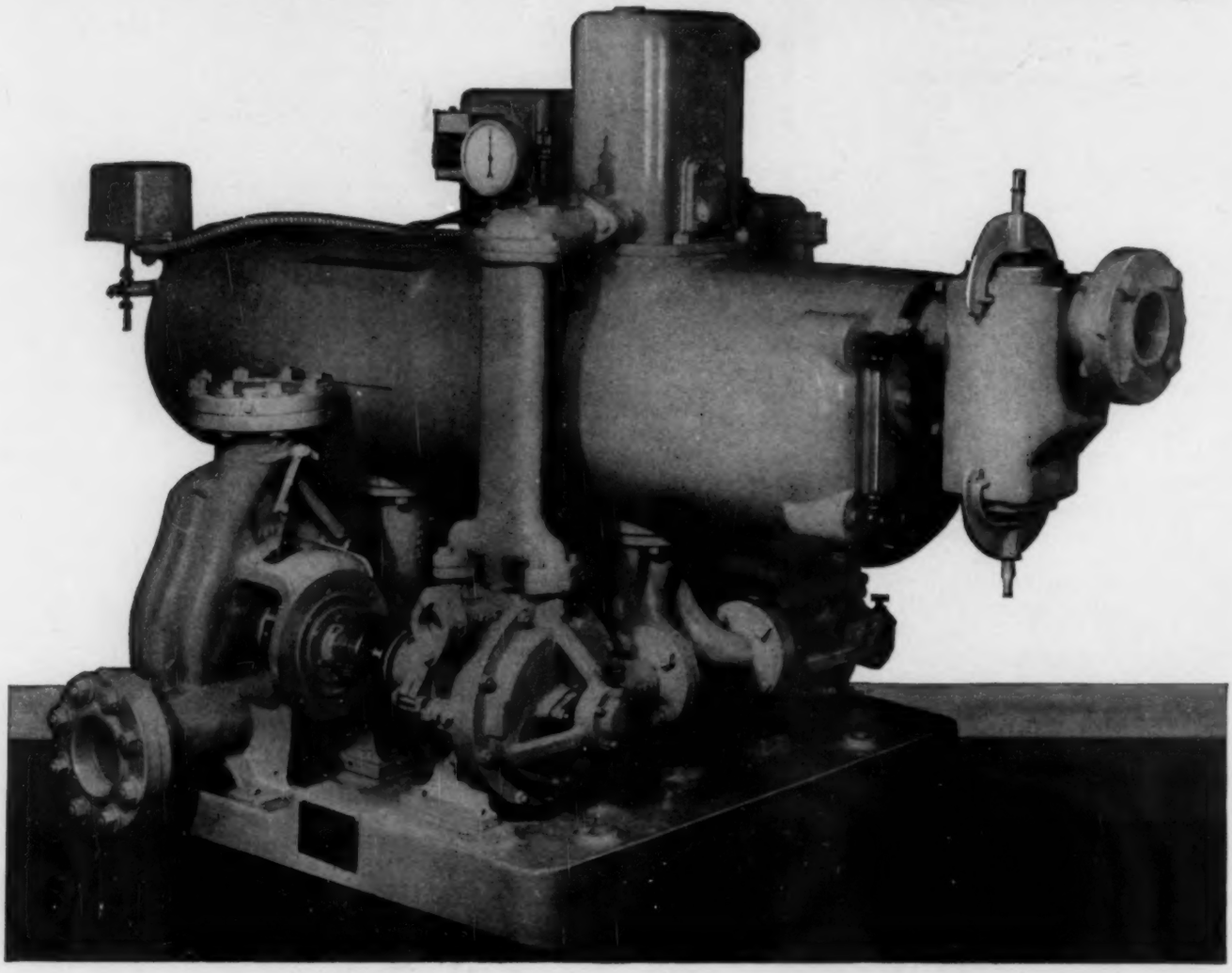

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Heating · Piping and Air Conditioning

Vol. 11, No. 1 • *January, 1939*

Air Conditioning in Industry

The Flour Mill

FLOUR milling—in simple terms—is a process of alternately grinding and sifting wheat. The ingredients of the finished product are wheat and air. Ever since the first miller ground his first measure of wheat, the wheat itself has been a subject of absorbing interest to him and his successors. He studies its quality, weight, size, degree of firmness, and its "looks" and "feel." There is a milling difference between hard and soft, between old and new, between wheat from one section and wheat from another section, and the miller must know what these differences are and how they affect his operation and his finished product. Contrasted with the close study that the wheat receives, the importance of the other ingredient, air, has had scant attention until the last decade or so.

That air is an ingredient may, at first thought, appear to be a matter for argument; on closer examination, all argument vanishes. The air has had intimate contact with the wheat at all stages of the milling process for centuries. There has never been a "shortage" of air in milling. What was not appreciated until recently is that the products of the wheat berry are among the most hygroscopic of materials, and that the air always affects these products—for good or bad, depending on the kind of air it happens to be.

Not only the relative humidity, but the temperature and cleanliness of air in the mill are important. Flour is a food. It should be made clean and kept clean. In its finished state, it is literally finer than dust, and atmospheric dust can mix with it all too readily. Temperature is a factor too, both for the obvious reason of the operator's comfort and for other reasons which have to do with the adjustment of the milling machinery and the rate of flow of the wheat products from one machine to the next.

*By George M. Sebree**

Flour milling is a process of alternately grinding and sifting wheat, and the ingredients of the finished product are wheat and air. Ever since the first miller ground his first measure of wheat, the wheat itself has been of absorbing interest to him and his successors. But it has been only within recent years that the other ingredient—air—has received much attention . . . Mr. Sebree is something of a specialist when it comes to air conditioning of flour mills, and he discusses the problem here and describes how air conditioning has been applied at two modern mills

Part of roll floor, Tri-State mill, showing ductwork

In the fall of '37, the Tri-State Milling Co., Rapid City, South Dakota, started construction of a new mill which was to be modern in every respect. Horner & Wyatt, consulting engineers, were employed to design the building, and the new storage elevator built at the same time. The engineers designed a mill building which gave flour mill air conditioning a chance to really go to work and get results. This building is 37 ft x 80 ft in plan, six floors with a total height of 80 ft, and is almost windowless. The extensive use of glass brick in the walls makes windows unnecessary for light, provides insulation against heat and moisture leakage, and was an important factor in making it possible to design what is a most graceful and beautiful industrial structure.

This mill has a capacity of 600 barrels of flour per 24 hours. It is one of three mills designed by Horner & Wyatt during the past three or four years, each for a different company, each of about the same daily capacity, but all in the middle west and all air conditioned. Another of these mills is the Nebraska Consolidated Mills Co. plant at Grand

*Western Air Conditioning Corp.

Island, Nebraska, completed in 1936. This mill uses substantially the same type of air conditioning system which was installed in the Tri-State mill, and the following discussion will apply in general to both of them.

Why Air Conditioning Is Needed

In addition to improving operation of a flour mill, there is another "dollars and cents" reason for air conditioning—the possible saving of from ½ of 1 per cent, up to a full 1 per cent, of the raw material, the wheat, passing through the mill. This percentage represents moisture lost through needless evaporation. Such evaporation, in fact, usually will adversely affect the quality of the flour manufactured, because flour needs a minimum moisture content for stability, and to have the best keeping, mixing, and baking properties. The percentage of saving may seem small, but in a 600 barrel mill, if 1 per cent can be saved, this means six barrels of flour per day *not lost*—and this six barrels of flour may be worth around $5.00 a barrel wholesale.

The Control of Evaporation

The best general purpose flour has a moisture content of between 13.5 and 14 per cent by weight, I believe. The wheat will enter the mill at any moisture content of from 16 down to 12 or even 10 per cent. Deficiencies in moisture content in the wheat can to some extent be made up by "tempering," or soaking in water. Therefore, the wheat can be made to enter the first break roll at a more or less uniform moisture content. But the moisture content of the flour at the end of the milling process was formerly a matter that was on the laps of the gods. Now and then this moisture content was just right, sometimes it was too high, but more frequently it was far too low—and there was nothing that could be done about it.

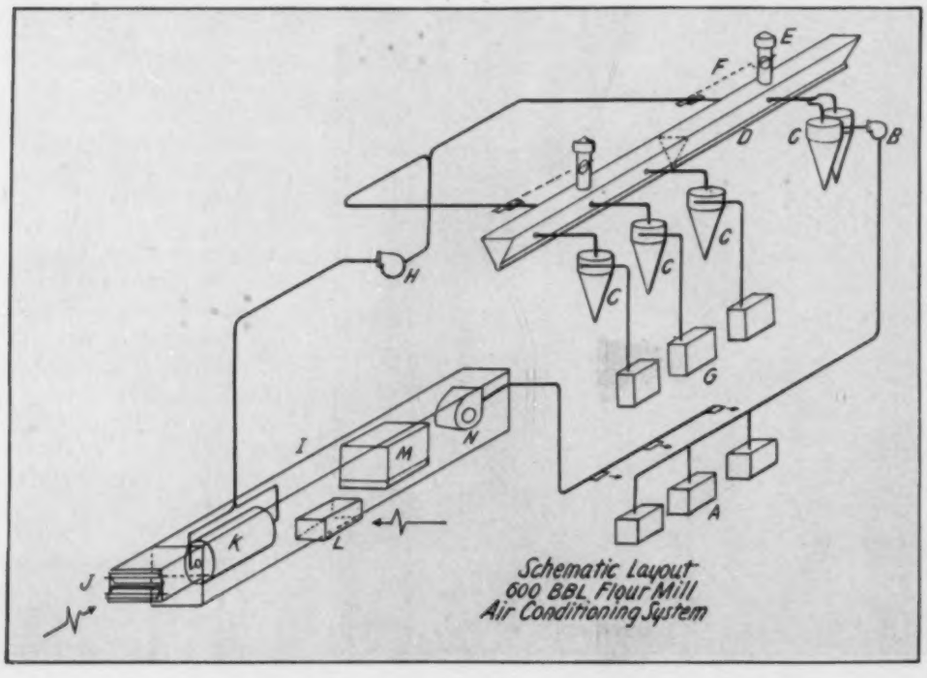
There is a limit to the water the wheat will absorb in the tempering process. If water is added above this limit, the wheat will "mush" in the rolls instead of breaking cleanly, and of course will not separate properly on the sieves. Water as a liquid cannot be added to the flour, for the result would be dough. Storing the flour in a humidified room offers little promise because of the slow rate of moisture absorption into the middle of the package after the outside layer of flour has absorbed even a little moisture. Few mills have space to store even their own production for more than a few days, yet thorough and uniform moisture absorption by packaged flour would require weeks and perhaps months.

The only way that flour can be delivered to the packaging machines at the desired moisture content is by the control of evaporation during the 20 to 30 minutes that the wheat products are in the process of being milled. This in turn means control of the relative humidity of the air in contact with them. In order to control the relative humidity, the temperature of the air must be controlled, and, as will be seen below, temperature control also brings its own independent advantages.

A relative humidity of 60 per cent is considered most satisfactory for milling, and this is the relative humidity that is maintained, except in winter, surrounding the roller mill machines in the Tri-State and Nebraska Consolidated plants. During extremely cold weather, even these well insulated buildings will not carry 60 per cent relative humidity without some condensation inside, and during the winter the relative humidity is reset to 40 per cent and compensated for by a change in the wheat tempering.

The central factor in flour mill air conditioning is that the mill stocks, the products of the wheat berry in smaller and finer sizes as these products pass through the rolls and sieves, tend to lose moisture if given an opportunity to do so. In an unconditioned mill, the low relative humidities that usually prevail will evaporate moisture from the stocks at any and every point in the mill. Perhaps a greater cause of moisture loss, particularly in the summer, is flash evaporation from the stocks during the brief contact with the hot rolls. A considerable amount of power is used to grind the stocks, and some of this power is converted into heat on and in the rolls. These rolls are good sized cylinders of cast iron and have a large heat holding capacity.

Of course, we could stop evaporation entirely, or even reverse the process to absorption of moisture by the mill stocks, by maintaining a high enough relative humidity. It is undesirable to stop



A, line of roller mills on third floor; B, roll suction fan; C, cyclone dust collectors; D, air collector trunk on ceiling of top floor; E, exhaust ventilator through roof; F, roll return and exhaust dampers; G, line of purifiers on fifth floor; H, return air suction fan on top floor; I, air conditioning equipment room on fourth floor; J, roll and purifier outside air dampers; K, stocking type dust collector; L, return air opening to fourth floor, with heating coil and hand damper; M, two stage air washer; N, main air supply fan to discharge grilles near ceiling of third (roll) floor.

evaporation completely, however, for a number of reasons, one of which is that in summer, with present roller mill equipment, there would probably be trouble in holding roll temperatures down to a reasonably warm point unless some evaporation were permitted. If we maintained the humidity at the equilibrium point for the moisture content of the entering mill stocks, we theoretically would have no evaporation. Equilibrium relative humidities and moisture contents of some hard winter wheat stocks have been determined by Dr. C. O. Swanson of the department of milling industry, Kansas State Agricultural College, and his associates, and reported in the *Northwestern Miller*, and are as follows::

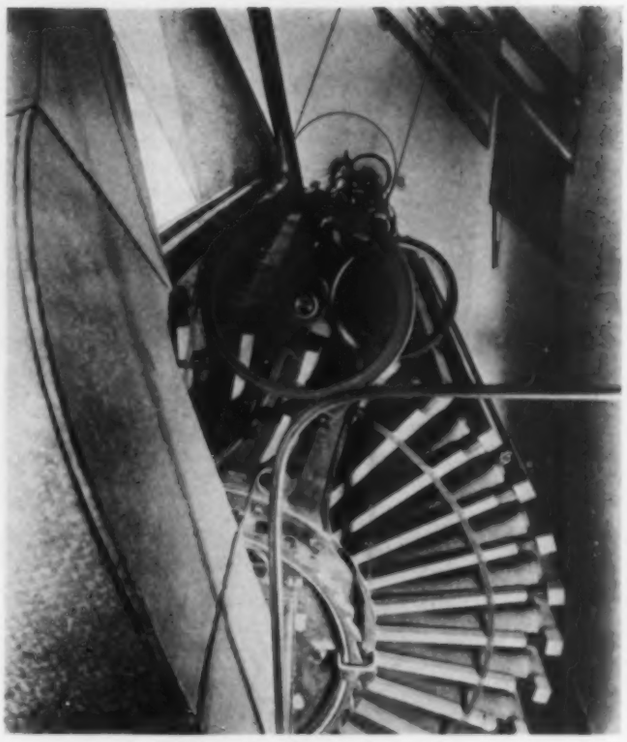
R. H. Per Cent	Per Cent Moisture in Stock				
	Wheat	First Break	Sizings	Flour	Bran
50	12.00	11.85	11.65	12.00	11.75
55	12.85	12.75	12.60	12.90	12.60
60	13.75	13.65	13.60	13.70	13.65
65	14.70	14.65	14.60	14.50	14.70
70	15.60	15.75	15.60	15.35	15.70
75	16.50	17.10	16.75	16.15	16.75

It will be noted from the table that flour with the desirable moisture content of 13.7 per cent moisture has an equilibrium relative humidity of 60 per cent, and this with wheat entering the mill at about the same moisture content. However, we must allow for some evaporation on the rolls, so the wheat is brought in at, say, 15.7 per cent moisture content instead of 13.7 per cent. If the mill stock could remain in the mill in intimate contact with the 60 per cent relative humidity air long enough, it would return to 13.7 per cent moisture content, no matter what its entering moisture content. But we cannot wait that long, for the stock must flow through the mill without delay. The best thing is to bring the stock in at the moisture content most favorable for the final moisture content desired in the flour.

Temperature a Factor

Closely connected with the relative humidity (in flour mills as everywhere else) is temperature. An ideal temperature for milling would be about 78 F the year 'round. This applies to the roll floor and may be allowed to vary somewhat on the other floors. In summer, this calls for cooling. In winter, some heating may be required. Note the word "may," because there is another source of heat in the mill, ready at hand if it is put to work. This is mechanical heat generated by the moving mill machinery. About 0.4 horsepower per barrel of daily capacity is required to drive the rolls, sifters, purifiers, conveyors, elevators, and other machinery in the mill proper, and the separators, scourers, etc., in the wheat cleaning department. In a 600 barrel mill, the very respectable total of $(600 \times 0.4 \times 2546) = 611{,}040$ Btu per hr might be available as waste heat if it could all be trapped and kept inside the building. Not all of it can be kept inside by any means, but enough can be salvaged to make a very real saving in fuel required to heat the mill.

As a general rule, neither the Tri-State mill nor the Nebraska Consolidated's Grand Island plant require heating by their steam plants until the outside temperature drops to around 30 deg. As a matter of fact, one of the principal functions of the outside air damper in the air conditioning system in these mills is to be always ready to swing open as required to *prevent overheating in winter*. And while we're on the subject of apparent paradoxes, an important duty of the steam heating system in these mills is to heat the building to a very high temperature in *summer*. When outside conditions permit, both mills are "fumigated" by bringing the inside temperature up to around 145 F and maintaining this temperature for several hours.



Part of stocking type dust collector, also return air damper in partition

Air Conditioning Must Start with Structure

It is important to remember that mill air conditioning must start with the building structure. It does little good to produce the desired conditions inside unless the building can hold these conditions. Both the Tri-State mill and the Nebraska Consolidated mill have brick and hollow tile walls, plastered inside, and have glass brick panels instead of windows. Rigid insulation is used on the roofs, and in general both buildings are well built to prevent heat and moisture transfer between inside and outside. In the winter time particularly, insulation is very necessary to satisfactory operation of the air conditioning system.

Back about the time of the World War when mill construction was booming, a definite type of building became almost standard. This was the "daylight mill," or the "sunshine plant"—words which spell many problems for the air conditioning engineer. Such a mill frequently consisted of a reinforced concrete frame, with outside walls having vast areas of single thickness glass. If it is proposed to install air conditioning in a building of this sort situated in a cold climate the best investment that could be made would be to insulate it. This would involve considerable rebuilding or revamping of the outside walls, but should be considered as part of the necessary expense of installing the system.

Saving part of the machinery heat makes the task of heating the mill in winter much easier. Yet in many mills, no attempt is made to save any of this byproduct

The Tri-State mill, Rapid City, South Dakota

heat. It is all exhausted outside including a very large part of the heat supplied by the heating system. With the rolls, purifiers, and sometimes the elevators, all requiring air under suction for ventilating, or separation of light from heavy stock by aspiration, these machines may exhaust all the air in the mill every 8 to 10 minutes. And in winter, there is nothing to take its place but cold air from outside.

A bad beating is taken in winter by the average mill which is situated in a cold climate. It frequently does not save any part of the heat from its own machinery, and is forced to use and heat large volumes of cold outside air. And its glass area and thin uninsulated outside walls add an unreasonable amount to the total heating load.

The Air Conditioning Cycle

While it was mentioned above that in general the same type of system was installed in the Tri-State mill and the Nebraska Consolidated mill, there are of course some differences in these systems, and from this point on the equipment in the Tri-State mill will be followed.

This mill has six floors, the first being used for storage and packing and the second for cereal milling. The third floor is the "roll floor," the heart of the mill, and it is here that the effect of the air conditioning system is concentrated. The fourth floor is used for general purposes and the air conditioning equipment room is situated there. The fifth floor contains purifiers and sifters, and the sixth floor is used for the cyclone dust collectors, blowers, and air ducts. The cleaning department is also on the five upper floors of the building, but on each floor is sealed off tightly from the milling section.

A good point to start with a description of the air conditioning cycle is the air exhausted from the milling machinery, this air being roughly half the total of 15,000 cfm delivered by the system. Of the approximately 7500 cfm exhausted from the milling machinery, about 40 per cent comes from the rolls. This air is usually 12 to 15 deg warmer than the surrounding air in the mill, the air being heated by the rolls themselves, which have been warmed by mechanical friction. The net moisture content of this air has also increased due to evaporation from the stock. This roll air is what is used to build up humidity and heat in the mill in winter. In summer, all of the roll air is exhausted outside, because cooling the outside air to take its place always constitutes a smaller load than cooling—and to some extent dehumidifying—the roll air.

Most of the remaining 60 per cent of exhaust air from the machines comes from the purifiers. Each purifier has its own fan, which draws air up through sieves on which the stock is being separated. When discharged from the purifiers, the air has picked up little or no heat. Its moisture content may have increased by a very slight amount due to some evaporation from the stock on the sieves. It is at almost the same condition as the air in the mill. Under automatic control, it is usually not exhausted outside in winter, but may be exhausted at other seasons as required to maintain inside conditions.

As will be seen from the schematic layout reproduced here, both roll air and purifier air is first discharged into the cyclone dust collectors, where the heavier and coarser dust is precipitated by gravity and centrifugal action. From the top of the cyclone dust collectors, the air passes to a collector trunk, a large duct usually built in triangle form with steeply sloping sides, and which has a partition to separate the two kinds of air. Additional dust is precipitated in this trunk and collected by a screw conveyor at the bottom.

Each main chamber of the collector trunk has an exhaust ventilator head extending through the roof of the mill to atmosphere. A heavy steel plate damper is installed in the throat of each ventilator. A return duct is brought out of the side of each chamber of the collector duct, and these ducts also have heavy steel plate dampers. The damper motor and control arrangement is such that when either exhaust damper or ventilator damper is open, its companion return damper is closed (or *vice versa*). And more than this, when either exhaust damper opens, an outside air damper of equivalent air capacity opens at the same time to admit the exact amount of outside air needed to prevent a pressure or vacuum condition in the mill.

Neither exhaust damper ever closes tightly, about 5 per cent of the air being given a chance to escape at any time. We cannot balance the amount of air delivered into the collector duct and the amount of air returned from this duct to the mill down to the last cfm. Therefore, we prefer to allow a little air to escape to the outside rather than to take a chance of pressure building up in the spouts and blowing dust out into the mill through spout inspection plates. It might be mentioned also, in passing, that a pressure or vacuum condition in the mill has a bad effect on the performance of the purifiers. The purifier separates mill stocks partly by aspiration, which involves a delicate balance between the suction of the purifier fan and the amount of air admitted under the sieves. This balance can be maintained only by maintaining constant pressures on the outside of the machine and in the discharge duct from the purifier fan.

The return ducts from the two chambers of the collector trunk join as soon as possible after passing the separate return dampers, and the single duct thus formed is brought to the suction of the main return fan. This fan is driven at a constant speed and may at times operate without any air supply at all, if both purifier and roll return dampers should happen to be closed at the same time. This fan has ample capacity to handle full volume of air from the rolls and purifier at the same time.

The main return fan discharges into a duct which is brought down from the sixth to the fourth floor, and which enters both ends of a stocking type dust collector, where practically all of the balance of the dust is removed from the air. This collector is situated in the air conditioning apparatus room. The stocking type dust collector itself is a large cloth filter, revolving slowly and stopping perhaps 40 times during each revolution. It is more or less self-cleaning, having an arrangement of rubber tipped hammers which give several hard thumps to each row of "stockings" when this row is momentarily stopped in the top position of the circle. At this position, this particular row of stockings is under suction, while all the other rows are under pressure of something upwards of 1 in. of water. The dust precipitated by this combination of thumping and suction is removed by a screw conveyor.

If our ventilator dampers are closed, and return dampers are open, we will now have about 7500 cfm of clean air issuing from the stocking type dust collector. This air is in the conditioned air stream—in fact, it is the conditioned air that in winter will maintain the mill building at the temperature and humidity for which the instruments are set. This applies, of course, if the outside temperature is somewhere between 30 and 70 F; if the outside temperature is below 30 F, it will be necessary to call on the mill's heating system for some help.

Additional Heating of the Air

Additional heat is imparted to the air, when needed, in this manner: The stocking type dust collector, and all the ductwork, etc., between it and the milling machinery, can supply only half the air handled by the entire system. Assuming that the outside air dampers are closed, as they normally would be during cold weather, the remaining 7500 cfm will be return air, taken directly through an inlet opening in the air-tight partition between the air conditioning equipment and the mill. In this opening is a two row steam blast coil made up of copper tubes and aluminum fins. A modulating type motorized steam valve admits steam to this coil as needed. This steam valve is controlled by a thermostat set at 76 F situated on the main instrument panel in the mill. The thermostat controlling the purifier dampers is set to maintain 78 F. If the outside temperature drops to the point when the mill cannot be maintained at 78 F by the heat from the machinery, the second thermostat controlling the steam valve comes into play when the mill temperature has dropped 2 deg.

Between the stocking type dust collector and the main fan is situated the two stage air washer which cools the air, and hence the mill, in summer. The first bank of sprays is normally connected to recirculate water. The pump motor for this bank is automatically controlled by a thermostat, and starts when the temperature in the mill rises above the thermostat setting. The second bank of sprays receives water from a well pump, delivering 100 gpm of 50 F water, which is the temperature of the water obtainable at Rapid City. The motor for this pump is automatically controlled by a hygrostat in the mill, cutting in as necessary if the inside humidity should rise above the hygrostat setting.

Grand Island plant of Nebraska Consolidated Mills Co.

When operated on the summer cycle, the automatic controls are so arranged that when the temperature begins to rise in the mill, the purifier ventilator and purifier outside air damper will open and the purifier return will close, in order to do as much cooling as possible with outside air. If the temperature continues to climb, another thermostat will start the recirculating pump forcing water through the first stage nozzles of the air washer. As both outside air dampers are open, the equipment will be handling a large proportion of outside air, and will do a considerable amount of cooling by evaporation.

As a matter of fact, in this mill in the Black Hills of South Dakota, evaporative cooling has been all that has been needed for a very large part of the time during the past summer. It has maintained the temperature desired, with sufficient but not excessive humidity. Should very high outside temperatures prevent this cycle working to advantage, the milling superintendent needs only to go to the panel board, flick a switch, and return the purifier air to the equipment, shutting off half the outside air. As a last resort, he might return the roll air also, shutting off practically all of the outside air, but the outside air would have to be exceptionally hot and humid before it would pay to operate with roll air returning to the mill.

The outside air intake is situated on the fourth floor 50 ft above the ground. This is a naturally dust free location, and in summer all the air entering the mill passes through the sprays and is washed. In the Black

[Concluded on page 19]

New Charts for Figuring

Friction Loss in Piping

IT is common practice to rely upon empirical formulas to determine the friction loss in pipes for computing the flow of liquids and gases; this is particularly true with water, air and steam. In general, empirical equations give excellent results within the test ranges from which they were derived, but usually they do not take into account the changing characteristics of the fluids due to temperature changes.

By E. A. Windham*

Mr. Windham is concerned in his daily work with computing the flow of various fluids in piping, and he developed the charts reproduced here in order to simplify the routine involved. . . . With the hope that they will prove useful to others who must solve similar problems, he has made them available to readers of HEATING, PIPING AND AIR CONDITIONING

The hydraulic engineer—dealing entirely with water within a limited temperature range—can quite safely use an empirical formula. In the same way, the ventilation engineer may safely use an empirical equation for determining the friction loss in ducts when he is considering air at or about standard conditions. It frequently occurs, however, that the friction loss in pipes for various fluids or gases over a wide range of temperatures must be computed, and I think it is generally agreed that it is unsafe to use empirical equations for this type of work. When this problem arises, the best solution (to my knowledge) is to utilize the Fanning equation:

$$\triangle P = \frac{CfSV^2L}{D}$$

where $\triangle P$ = friction loss in pipe;
C = a constant;
f = friction coefficient;
S = pecific gravity (62.4 lb per cu ft = 1.0);
V = elocity;
L = ength of pipe;
D = nternal diameter.

The friction coefficient f is a function of the Reynolds number Re.

$$Re = \frac{KDVS}{Z}$$

where Z = viscosity;
K = a constant.

While this formula takes into account all of the known factors that affect friction loss in clean pipes, it is rather cumbersome to use—which probably accounts for its not being as widely used as would be expected. I have utilized a method of simplifying the application of this equation which has been of great aid to me and I hope it will be of assistance to others.

*Engineering Dept., Surface Combustion Corp.

The friction coefficient is a function of the Reynolds number, or

$$Re = \frac{KDVS}{Z} \propto f$$

Therefore, for any given value of (DS/Z) the coefficient f is a function of the velocity V. Also

$$\frac{\triangle PD}{SL} = CfV^2$$

which means that for any given length of pipe $(\triangle PD/S) \propto fV^2$. Therefore, for a given length of pipe it is possible to plot the term $(\triangle PD/S)$ against the velocity V for any given value of (DS/Z).

The two charts show the values of $(\triangle PD/S)$ for various velocities and for various values of (DS/Z). The second chart is merely an extension of the first and for convenience $\triangle P$ is taken to be the pressure drop in 100 ft of pipe, in feet of water. The (DS/Z) lines have two slopes, one representing streamline flow and the other being the turbulent flow region. The dotted portion indicates the range between streamline and turbulent flow which may be either type of flow. However, it is considered as turbulent flow since this will give the larger friction loss and therefore is conservative.

An example of the use of this chart follows:

Determine the pressure drop due to friction in 100 ft of 3 in. pipe when handling 200 gpm of a brine having a specific gravity of 1.31 and a viscosity of 10 centipoises ($D = 3.068$ in., $S = 1.31$, $V = 520$ fpm, $Z = 10$).

$$\frac{DS}{Z} = \frac{3.068 \times 1.31}{10} = 0.402$$

From chart, $(\triangle PD/S) = 43$. Therefore,

$$\triangle P = \frac{43 \times 1.31}{3.068} = 18.4 \text{ ft of water per 100 ft of pipe.}$$

You will note that if the value $(\triangle PD/S) = 43$ is divided by the pipe diameter D the result would be $(\triangle P/S)$, which would be the friction loss in feet of the liquid in question per 100 ft of pipe. This is very useful in handling pump problems.

These curves also apply to gases. As an example, let us determine the friction loss in 100 ft of a 12 in. round duct handling air at 1400 fpm at 170 F. ($Z = 0.0202$; $D = 12$; $S = 0.063 \div 62.4 = 0.00101$; $V = 1400$.)

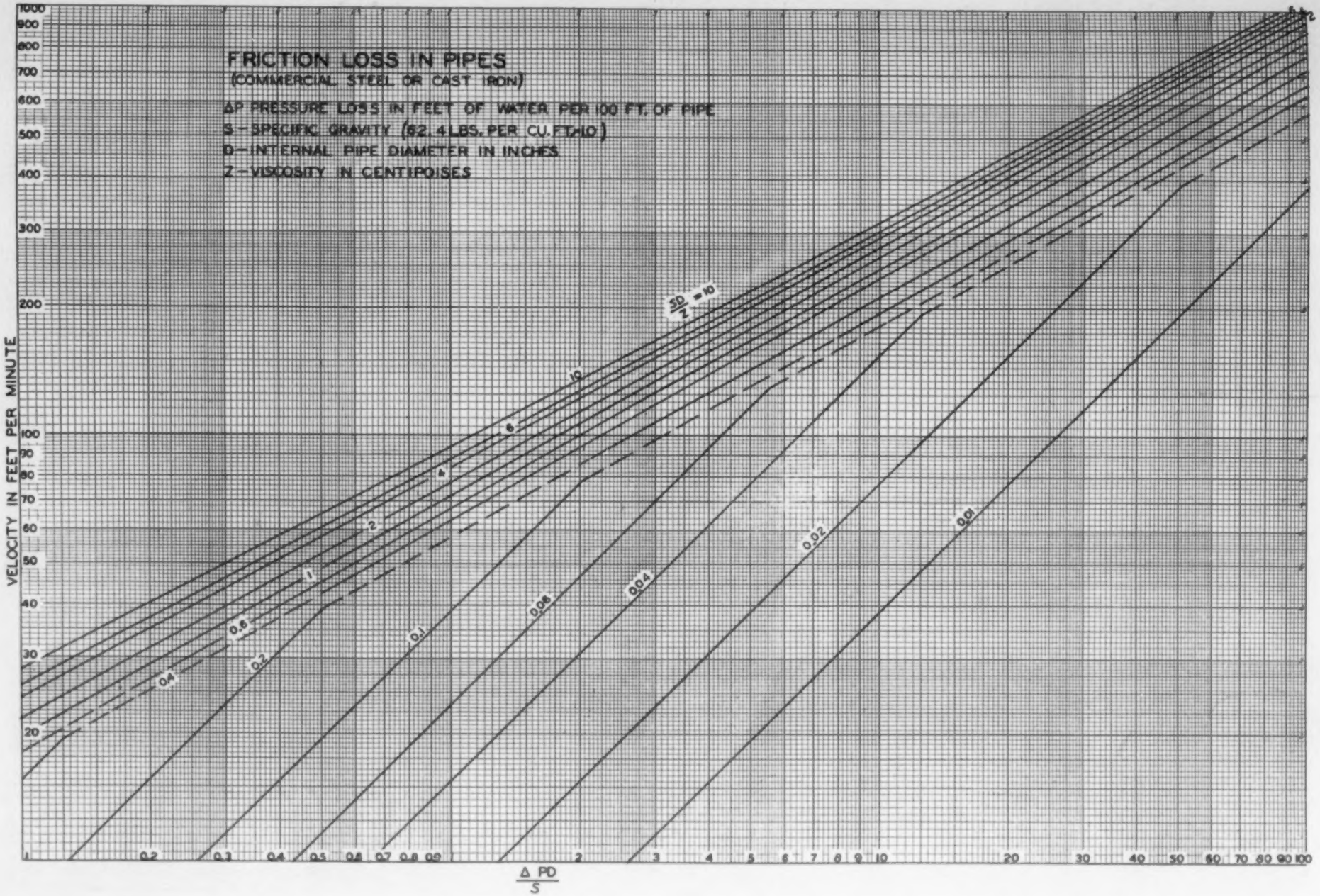
$$\frac{DS}{Z} = 0.6$$

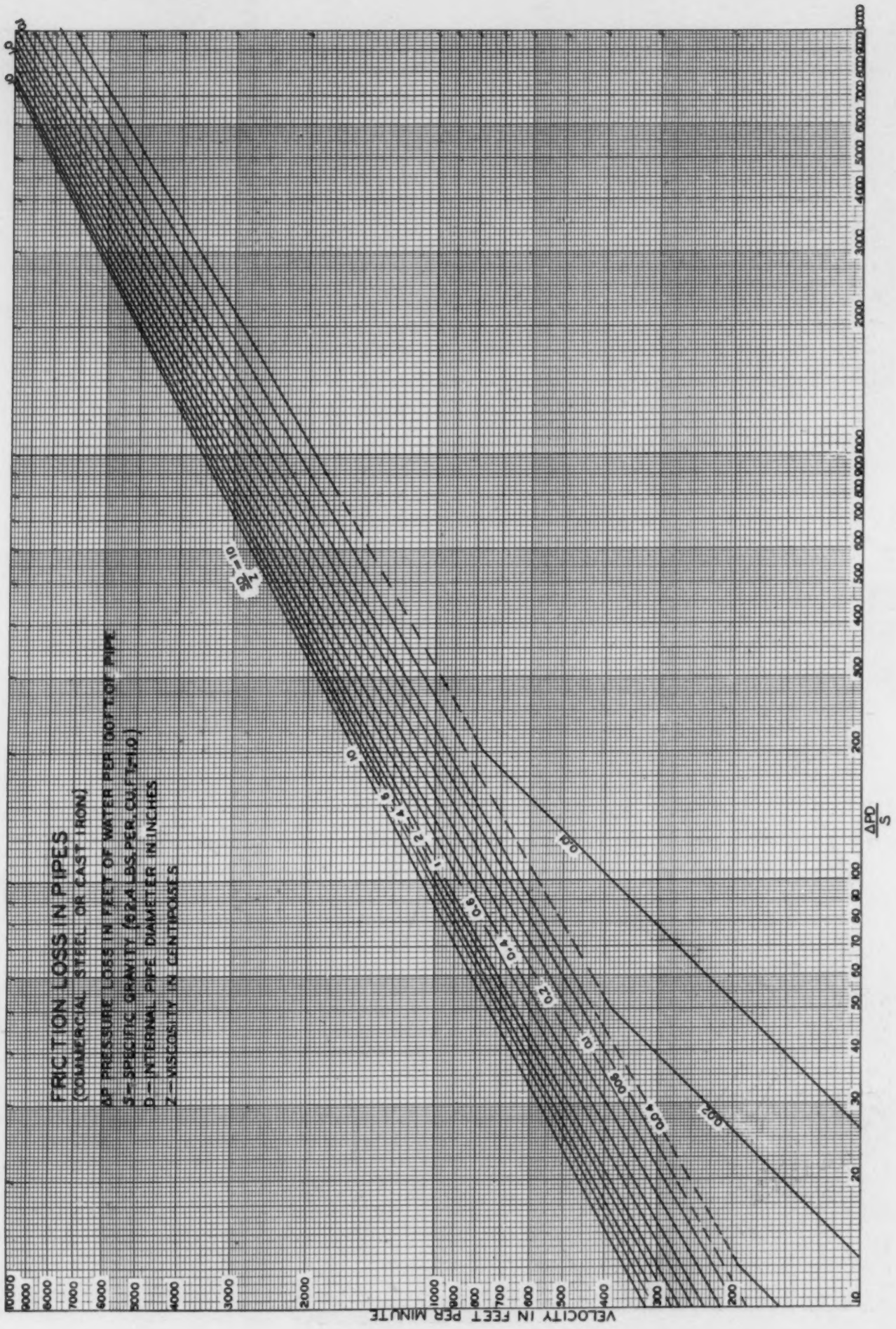
From chart

$$\frac{\triangle PD}{S} = 234$$

$$\triangle P = \frac{234 \times 0.00101}{12} = 0.0197 \text{ ft of water per 100 ft}$$

$$= 0.237 \text{ in. of water per 100 ft}$$


FRICTION LOSS IN PIPES
(COMMERCIAL STEEL OR CAST IRON)
ΔP PRESSURE LOSS IN FEET OF WATER PER 100 FT. OF PIPE
S – SPECIFIC GRAVITY (62.4 LBS. PER CU. FT.=1.0)
D – INTERNAL PIPE DIAMETER IN INCHES
Z – VISCOSITY IN CENTIPOISES
VELOCITY IN FEET PER MINUTE
ΔPD/S
SD/Z = 10
10
6
4
2
1
0.8
0.6
0.4
0.2
0.1
0.08
0.04
0.02
0.01


FRICTION LOSS IN PIPES
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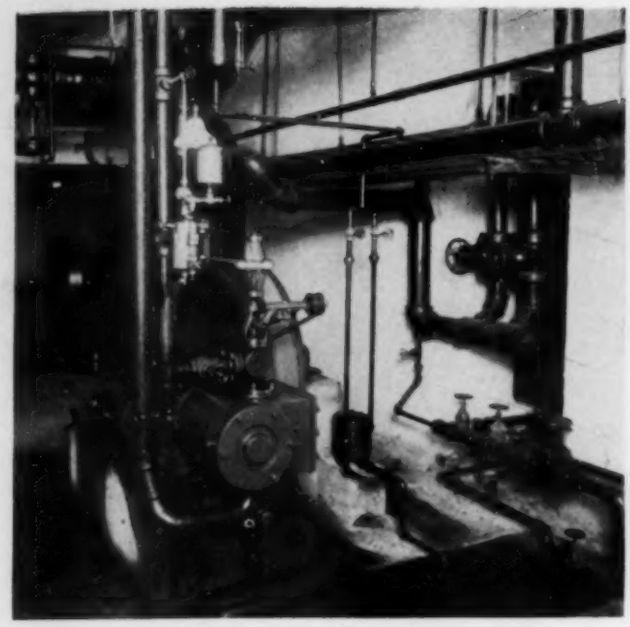
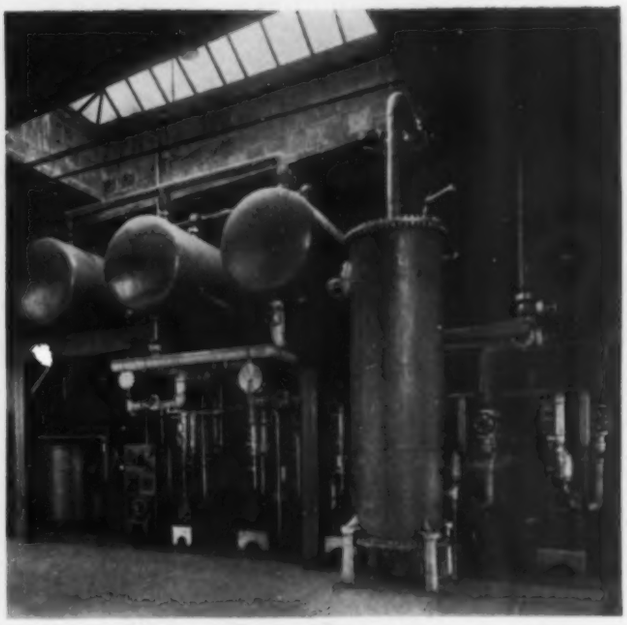
Z—VISCOSITY IN CENTIPOISES
VELOCITY IN FEET PER MINUTE
ΔPD/S



Engine driving ventilating fan



The pump room at a high school

Organized Maintenance Procedure Cuts Fuel Consumption for School Heating

An interview with John Howatt*

Careful organization for maintenance and operation of the heating and ventilating services in Chicago's 400 school buildings has resulted in a steady reduction in fuel consumption. It is estimated that, over the past five years, fuel for heating high schools has been cut about 12 per cent and for elementary buildings, about 5 per cent. These savings represent important money, for the school system burns some 180,000 tons of coal annually. . . . Mr. Howatt—who completed last month 25 years of service as chief engineer of the Board of Education—tells here the maintenance routines which have achieved these economies, and describes general policies on plant operation, maintenance, repairs, replacements, etc.

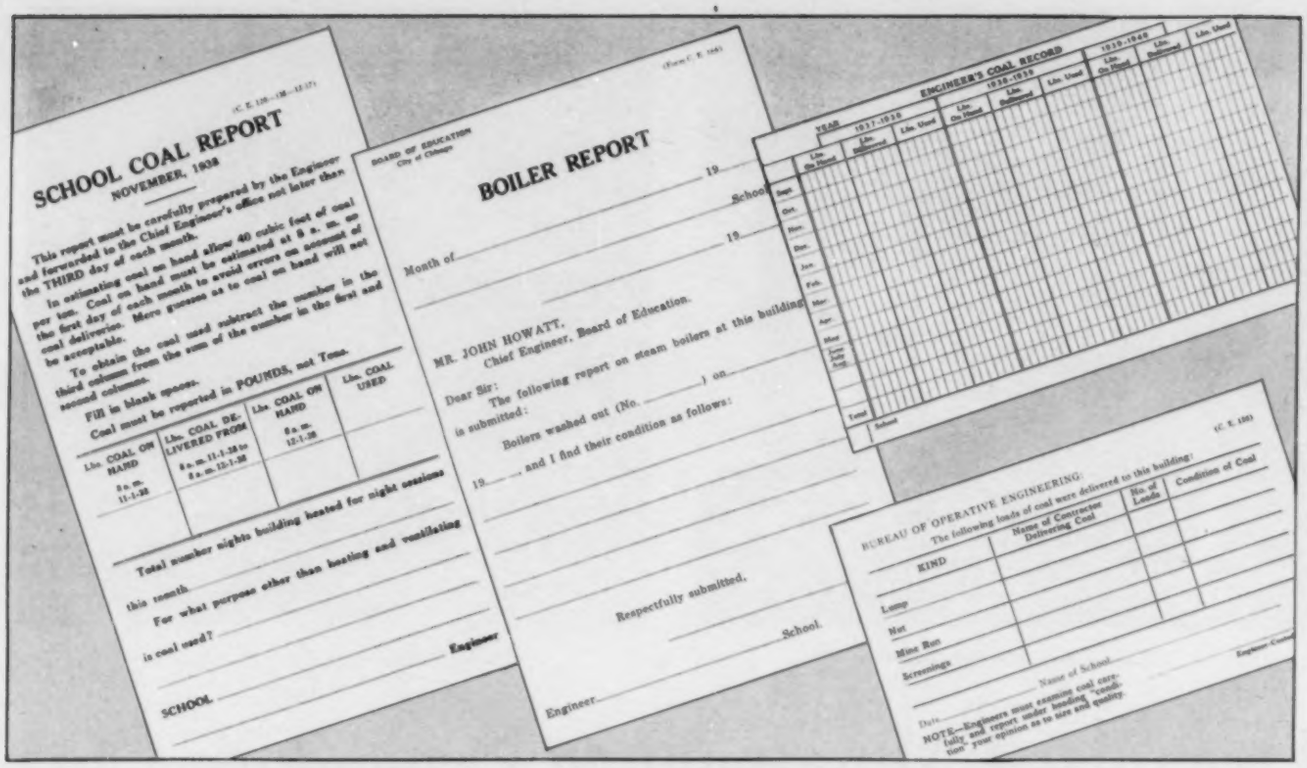
DESIGN, operation and maintenance of the heating and ventilating plants and the mechanical and electrical equipment of Chicago's public schools are under one control—that of the chief engineer of the Board of Education. Because of the great number of buildings and the combined size of the heating and ventilating installations, the system of operation and maintenance has been carefully worked out so that efficient and satisfactory service may be assured.

An idea of the size of system may be had from the fact that there is a total of 134,272 rated horsepower of boilers in the 400 buildings under the supervision of this department. One hundred and eighty thousand tons of coal are burned yearly to provide heat during school hours for half a million pupils and teachers. Eight hundred and thirty-three (770 high pressure and 63 low pressure) boilers of nearly every type — fire box, Scotch marine, shipping board, horizontal return tubular, and water tube—are in operation. Stokers of various types are used to fire these boilers, and many of them are hand-fired. About 20 small buildings have warm air fan furnace heating systems. For fuel, a good grade of Southern Illinois coal has been used for the last 10 years in all buildings but a few, which burn oil because of special considerations. To meet the ventilation requirements of the system, some 15,000,000 cu ft of outdoor air is introduced and distributed every minute of every school day.

The largest boiler plants are in the Teachers college (formerly the Normal school), where there are six 450 hp water tube boilers, and in the Lane technical high school, where there are four 600 hp water tube boilers. The sizes of the boiler plants range down to single 90 hp hand-fired fire box boilers. The size of the buildings ranges from 21,000 to 694,000 sq ft of heated and ventilated floor area.

Experience is the guide used in selecting equipment for the school plants. Comprehensive performance and maintenance records for the various buildings are carefully kept. These records throw abundant light on the relative value in actual service of various makes and types of equipment. If a boiler plant is needed for a new building, or an old boiler plant has to be replaced, it can be accurately

*Chief Engineer, Chicago Board of Education. Member of Board of Consulting and Contributing Editors.



Some of the forms used for boiler and coal reports

predetermined, from the records of other similar plants, what type of boilers, firing equipment and auxiliaries will give the best results and the greatest ultimate economy for the particular installation. The records show, for example, the cost of brickwork and repairs with various types of stokers on comparable installations, and the amount of coal (on the degree day basis) used with various types of boilers to heat each thousand square feet of floor space in buildings of similar size.

Accessories for the boiler plants have to be purchased with care and caution. The number of plant operating hours per year in schools is much less than in hotels or factories. Some of the auxiliary equipment which will repay its cost in savings in certain other establishments will hardly pay for itself within its useful life in a school boiler plant. It has been found that in school systems the simplest equipment is usually the best. Equipment that has been given a fair trial and failed, or whose maintenance cost is high, is not purchased again.

The heating peak in the city's school buildings is from 6:00 to 9:00 a. m., tapering down from 9:00 to 11:00 a. m. and carrying on at half peak from 11:00 a. m. to 4:00 p. m.—followed by 14 hours of inactivity. Heating equipment must be installed which will carry the peak, and also the lighter loads, efficiently and economically.

The criterion used to determine and compare heating costs is the amount of coal burned to heat each 1000 sq ft of floor space. The area and the cubic content bear a fixed relation to each other because room ceiling heights are uniform throughout the system. This figure, which is reported and recorded every month for each building, is of far greater practical value than reports of efficiency, stack temperature, evaporation and CO_2, as it is a composite report. This figure varies widely in the 400 buildings, due to structural conditions, boiler plant age, habits, and usage, as well as operating efficiency.

Boilers are kept in service for as many years as possible. A boiler is not replaced until it is on the verge of condemnation, or its physical condition is such that it is beyond the point of economic rehabilitation. Some of the boilers are delivering remarkably low cost heat after nearly 30 years in service. The average length of service for the system's boilers is between 24 and 25 years. The care with which inspections and repairs are made is evidenced by the fact that there has not been an explosion or serious boiler accident in the school system for 40 years.

How Maintenance Is Organized

The factors which make this record possible are careful selection of equipment, correct installation, supervised operation and adequate maintenance—of which the last is by no means the least important.

For the adequate maintenance of 400 heating plants operated under one head, proper financial provision and staff organization are prime essentials. Approximately $500,000 a year is set up in the Board of Education's budget for maintenance, major and minor repairs, and rehabilitation of mechanical equipment. As this total must not be exceeded, care is taken to spend somewhat less than the full amount during the year. This amount does not include a fund of $200,000 (a definite appropriation) for the replacement of boilers and plant equipment.

Under the chief engineer are six supervising engineers. The city is divided into six districts, each containing about 65 school buildings. Each supervising engineer has general charge of the operation and maintenance of the boiler plants and other mechanical equipment in the schools in his district. In each school the heating plant is operated by or under the direction of the engineer custodian.

Each supervising engineer is required to visit every building in his district at least once a month the year around and to make such additional visits as may be necessary. All are required to spend two afternoons a week in the central office and take up matters pertaining to their districts with the chief engineer. The supervising engineers' reports cover efficiency of operating crews, condition in which mechanical plant is kept, their own inspections of boiler and equipment, and recommendations for repairs and maintenance.

During the operating months the engineer custodians are required to blow down twice daily; blow soot twice daily, if the plant has soot blowers (if this is done by hand, the tubes must be punched once a week); empty boilers and wash out once a month; inspect boilers for pitting and scale; and to make a monthly report on condition of boilers. It is part of the duties of the supervising engineers to see that these practices are faithfully observed.

During the visits of the supervising engineer, the engineer custodian discusses with him needed repairs and maintenance. A decision is made between them whether the matter is urgent or if it can wait until summer. In the latter case, the repairs are included in the list of summer work for that building.

If it is decided that the repairs should be made at once, the engineer custodian issues a requisition, which, after being approved by the supervising engineer, is set up with an approximate cost estimate for the chief engineer's approval. When OK'd by the chief engineer, an order is promptly issued to a contractor, if the amount involved does not exceed $200. If the amount is more than $200, competitive bids are asked for, in accordance with the customary practice on city work.

Summer Work

When the school boiler plants are shut down in June, the boilers are not allowed to remain through the summer in a dirty condition while soot breeds corrosion and shortens the lives of the boilers, as is the case in too many heating plants. At shutting down time, the engineer custodian is required promptly to drain and wash boilers; wash all tubes thoroughly; remove all soot; and remove hand hole plates.

In early September, before the schools open, the engineer custodians are required to replace hand hole and manhole plates; put in new gaskets; fill and test the boilers; and to get everything ready for operation.

Brick repairs and other boiler plant repairs of a major nature are made during the summer, for thoroughness of repairs in summer minimizes the emergency repairs which have to be made during the heating season. Likewise, timely small repairs, properly made, obviate more expensive jobs later. Maintenance of piping, radiation, ventilating apparatus and all other mechanical equipment in the school buildings is likewise under the direction of the supervising engineers. Emergency repairs are made weekends.

The city boiler inspector visits each school building yearly, between March and September, and issues the customary certificate of inspection. As mentioned before, it is the policy of the school system not to replace a boiler until it is about to be condemned or is past repair. Sometimes, with the approval of the city boiler inspector, it is possible to keep an old boiler in service for a number of additional years by reducing the operating pressure. The maximum pressure for which boilers in the school buildings are designed is about 125 lb, and such pressures are carried only in the plants where steam is used for driving pumps, stokers, etc.

The policy with regard to replacement makes it essential, in the interest of economy, to keep all boilers in as efficient operating condition as is humanly possible. Special attention must be given the older boiler plants. This is where supervised operation and maintenance prove their worth.

The following tables show the coal consumption in tons for heating 1000 sq ft of floor area for the past six years, with the degree days (for the eight months, October 1 to May 31) given for purposes of comparison.

For All High School Buildings

Year	Coal Consumption	Degree Days
1932-33	4.8	5972
1933-34	4.7	6039
1934-35	4.5	6043
1935-36	4.7	6784
1936-37	4.3	6219
1937-38	4.1	5866

For All Elementary School Buildings

Year	Coal Consumption	Degree Days
1932-33	4.1	5972
1933-34	4.1	6039
1934-35	3.9	6043
1935-36	4.3	6784
1936-37	4.0	6219
1937-38	3.8	5866

The larger coal consumption in the high school buildings is due mainly to the fact that they are used and heated more hours per year than the elementary school

Boilers and chain grates in one of the high schools

buildings, as night classes are conducted in many of the high schools and they are utilized by outside organizations for various activities.

The comparison might be carried further by ascertaining the ratio of coal consumption in lb to degree days, per 1000 sq ft of area. For the school year 1932-33 this ratio for high schools would be 9600 lb ÷ 5972 = 1.6. For the school year 1937-38 the figure would be 8200 ÷ 5866 = 1.4.

A comparison of the fuel consumption figure shows that, under the present system, more economical heating is being attained as the years roll by. The same quality of coal has been burned during the entire period. A comparable heating job is being done with about 12 per cent less coal than was burned five years ago.

The same ratios for the elementary schools are 1.37 for 1932-33, and 1.3 for 1937-38, showing an economy of about 5 per cent. These percentage savings represent important money on a yearly fuel bill as high as that of the school system.

Readers Ask Information on

Underground Steam Line....Heating a Cordage Mill

Answered by Samuel R. Lewis*

A READER of H. P. & A. C. asks how condensate in an underground steam line can be best disposed of when the flow of steam is uphill at a slope of 10 ft in a hundred (steam at 75 lb and the distance traveled 600 ft):

In a general way an upgrade away from the boilers is desirable, rather than otherwise, provided that the main is not too small to prevent the condensate from working back against the steam flow, since it tends to prevent blowing out the remote end with a slug of water at high velocity when steam is turned on too suddenly.

I have designed many installations with a steep uphill slope and have had no trouble. I would provide a drain pocket full pipe size and several feet deep at the inlet end, from which the float trap releases the water. I would provide a similar pocket at the upper end of the line, several feet beyond the tee which looks up to remove the steam—this pocket also serving as a cushion tank to receive any possible impact from blown up condensate.

It is wise to use extra heavy fittings on such a line. If there is a high pressure return paralleling the main an intermediate condensate drainage pocket and trap would be helpful, say at every 200 ft, or at the intermediate expansion joints. If the parallel return main is at low pressure it will be necessary to install flash tanks for allowing the condensate to re-evaporate, as much of it will, due to superheat at the reduced pressure, before trying to drain into the low pressure return main. The high pressure trap drains into the flash tank and the latter drains the condensate residue through a second float trap into the low pressure return main.

The reader also asks what is the minimum slope of an underground steam main, 2400 ft long (steam entering at 100 lb and leaving at 50 lb with steam and condensate flowing in the same direction):

I prefer to have such a line pitch up away from the boilers than have it level, and I prefer to have the main pitch up or be level than have it pitch downward away from the boilers. This opinion is based on actual experience due to too hasty turning on of the steam, following which the remote end of a down-pitching line usually is broken loose due to simultaneous arrival there of steam and a large body of condensate.

If the line pitches up away from the boilers the original air in the pipe, which can escape only slowly, forms a cushion, and as the main warms up slowly the condensate drains back slowly and much of it tends to be re-evaporated due to superheat resulting from pressure reduction on account of friction.

Incidentally, a pressure reduction by friction only of about 2 lb per 100 ft run, as suggested in the reader's question, seems rather high.

Heating a Cordage Mill

Another correspondent wishes information concerning the heating of a cordage mill where cotton spinning machines induce large quantities of lint to float in the air and to cling to walls, rafters, windows, and the like. He wants to know if an air washer and steam coil system, with duct distribution, can be used:

Indirect heating with central fans, ducts and air washers has been used successfully for a great many years throughout the textile regions in the East. The higher relative humidity possible with such a system tends to reduce the floating lint and adequate screening and maintenance of the screens in the spray water circuit will prevent too much trouble from clogging of the spray heads.

It is desirable to design the eliminator plates with removable tops so that lint collecting on them can be washed down or scraped off. The water screens should be extra large and easily brushed off to a lint catcher. It is practicable in many cases to use 100 per cent recirculation of the air, with a great saving in cost of heating.

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

Performance of Coils for Dehumidifying Air

By William Goodman*

New and simple formulas are developed by the author for computing the final dry and wet bulb temperatures of the air leaving a coil. Particularly noteworthy is the fact that the final wet bulb depression of the air is shown to depend upon the initial wet bulb depression of the air for both direct expansion and counterflow coils. This makes computation of the final dry bulb temperature the work of a moment

THE surface temperatures of direct expansion coils, the "condition curve" for such coils, and the "boundary" dry bulb temperature of direct expansion coils were the topics discussed in Part 2. This month, the discussion of the boundary dry bulb temperature is completed, after which the important subject of counterflow coils is treated.

For a given coil and a given refrigerant temperature, Equation 13 can be used to find the boundary dry bulb temperature corresponding to different initial dew point temperatures. Plotting these boundary dry bulb temperatures and dew point temperatures on the psychrometric chart for different refrigerant temperatures results in a set of curves similar to those in Fig. 7. These curves of constant refrigerant temperature are of considerable value because they show at a glance the conditions under which the coil will be entirely wet, or partially wet and partially dry. If the initial condition of the air is represented by a point to the *left* of the curve for the given refrigerant temperature, the entire surface will be wet. On the other hand, if the initial condition of the air is represented by a point to the right of the temperature curve, a portion of the surface will be dry. For example if the refrigerant temperature is 50 deg, and the initial condition of the air is represented by point *1*, part of the surface will be dry.

Curves like those of Fig. 7 can also be used to determine the boundary dry bulb temperature. If the given temperature of the refrigerant is 50 deg, the air initially in the state represented by *1* must be cooled by the dry surface along a horizontal line of constant absolute humidity (moisture content) to point *B* on the curve for 50 deg refrigerant temperature before dehumidification can begin. On the other hand, if for any reason the possibility of dry surface occurring on the coil is to be eliminated, the refrigerant temperature must be lowered. The point to which the temperature of the refrigerant must be lowered can be read from the curve passing through point *1* of Fig. 7.

If a part of the coil is dry while the balance is wet, then the single coil must be treated as though it were two separate coils through which the air flows in series—the first coil being dry and the second wet. The same formula cannot be used for calculating the cooling capacities of dry and wet coils. If, after computing the boundary dry bulb temperature by means of Equation 13, a part of the coil is found to be dry, then the area of this dry surface must be found. The area can be found because the initial dry bulb temperature of the air entering the coil and the boundary dry bulb temperature, that is the final dry bulb temperature of the air leaving the dry part of the coil, are known. The area of dry surface required to cool the air from t_1 to t_B can be found by means of the following ordinary mean temperature difference formula using dry bulb temperatures instead of enthalpy.

$$H_D = A_D U_D D_M \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [17]$$

In this formula D_M is the mean temperature difference based on the initial and final dry bulb temperatures of the air and the refrigerant temperature. U_D is computed from the film coefficients by means of Equation 2. H_D is the amount of sensible heat removed in cooling the air from t_1 to t_B. The following example illustrates the method of finding the area of dry surface required to cool the air to the boundary dry bulb temperature.

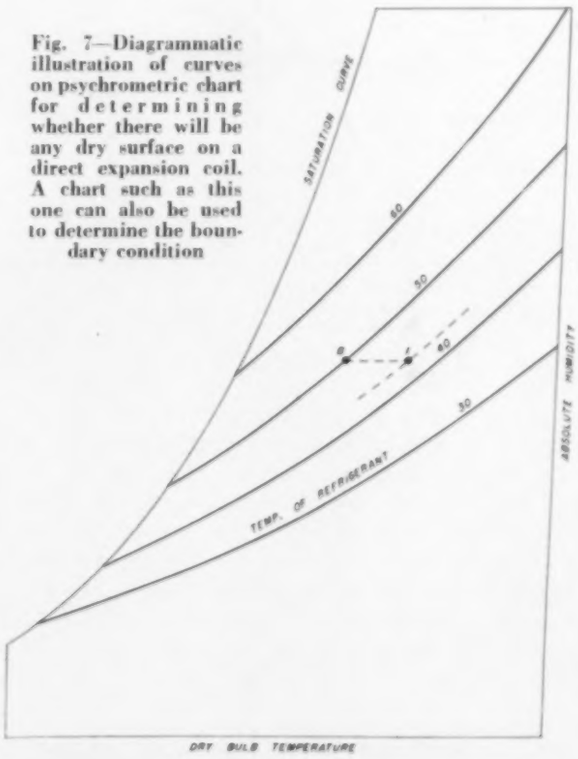


Fig. 7—Diagrammatic illustration of curves on psychrometric chart for determining whether there will be any dry surface on a direct expansion coil. A chart such as this one can also be used to determine the boundary condition

*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 3. Part 1 was published in November, 1938, pp. 697-701 and 707; Part 2 was published in December, 1938, pp. 777-781.

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Example 8: Take the conditions of Example 7 and find the area of dry surface required to cool the air to the boundary dry bulb temperature if the quantity of air to be cooled is 20,000 cfm.

Solution: From Example 7, the boundary dry bulb temperature is 85.2 deg. Equation 17 is used to find the area of dry surface required to cool the air from 98 to 85.2 deg. To find the mean temperature difference,

$$D_L = 98 - 42 = 56 \text{ deg}$$
$$D_B = 85.2 - 42 = 43.2 \text{ deg}$$
$$\frac{D_B}{D_L} = \frac{43.2}{56} = 0.771$$

Referring to Table 2 for $\frac{D_B}{D_L} = 0.771$, $\frac{D_M}{D_L} = 0.880$.

Therefore, $D_M = 0.880 \times 56 = 49.3$ deg.

From Example 1, $U_D = 7.14$.

$$G = 4.5 \times 20{,}000 \text{ cfm} = 90{,}000 \text{ lb per hr.}$$

Sensible heat removed in cooling the air from 98 degrees to 85.2 deg is $H_D = 0.243 \times 90{,}000\ (98 - 85.2) = 279{,}900$ Btu per hr.

Using Equation 17, the area of dry surface is

$$A_D = \frac{279{,}900}{7.14 \times 49.3} = 795.2 \text{ sq ft.}$$

If the final dry and wet bulb temperatures of the air leaving the coil of Example 7 must be found, only the area of the wetted portion of the coil should be used. Furthermore, the wet bulb temperature of the air entering the wetted portion of the coil will be lower than the initial wet bulb temperature of the air because of the sensible heat removed in cooling the air from the initial dry bulb temperature to the boundary dry bulb temperature. The temperatures and enthalpies at various points in a coil that is partly wet and partly dry are illustrated in Fig. 8. Strictly speaking, actual curves of temperature and of enthalpy would be difficult, if not impossible, to draw on one chart as in Fig. 8. Nevertheless a freehand sketch of such curves with the appropriate temperatures marked at the proper points makes an example like the following one easier to follow.

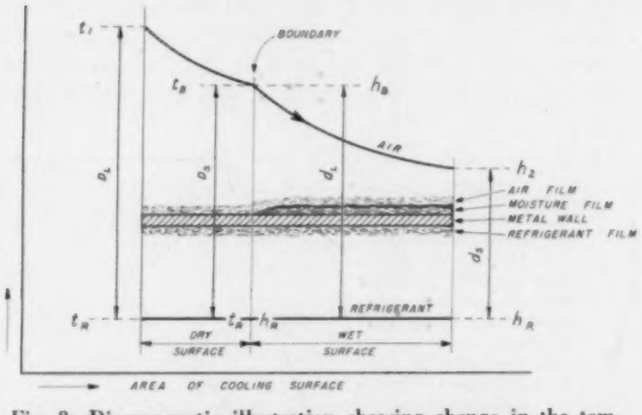


Fig. 8—Diagrammatic illustration showing change in the temperature and enthalpy of the air as it flows over a direct expansion coil that is partly dry

Example 9: Take the conditions of Examples 7 and 8. If the total area of the coil is 2000 sq ft, find: (a) the final wet bulb temperature of the air leaving the coil; (b) the final dry bulb temperature of the air leaving the coil.

Solution:

(a) $A_W = A_T - A_D$

$= 2000 - 795.2 = 1204.8$ sq ft of wet surface.

If the air is cooled from 98 deg to 85.2 deg at a constant dew point temperature of 59.5 as found in Example 7, the boundary wet bulb temperature as found from a psychrometric chart is 68.1 deg. (The boundary wet bulb temperature can also be computed by subtracting from the initial enthalpy, the sensible heat removed in cooling the air to the boundary dry bulb temperature.) From Table 4, for $t'_B = 68.1$ deg, $h_B = 32.45$ Btu; and for $t_B = 42$ deg, $h_B = 16.15$ Btu. In this case, $d_L = 32.45 - 16.15 = 16.30$ Btu.

In order to find the final wet bulb temperature of the air, the value of M must first be found.

From Example 1, $U_W = 19.9$.

By Equation 7, $M = \frac{1204.8 \times 19.9}{90{,}000} = 0.266$.

Referring to Table 3, for $M = 0.266$, $\frac{d_B}{d_L} = 0.766$.

$$d_B = 0.766 \times 16.30 = 12.5 \text{ Btu.}$$

From Equation 6, $h_2 = h_B + d_B = 16.15 + 12.5 = 28.65$ Btu.

From Table 4, for $h_2 = 28.65$, $t'_2 = 63.2$ deg, final wet bulb temperature of air leaving the coil.

(*Note*—The above method of finding the final wet bulb temperature can be used instead of the method illustrated in Example 4).

(b) The initial dry bulb temperature of the air entering the wetted portion of the coil is the boundary dry bulb temperature, $t_B = 85.2$ deg. Also for a boundary wet bulb of 68.1 deg, the enthalpy $h_B = 32.45$ Btu. In finding t_2, it is important to remember that in this case the boundary dry and wet bulb temperatures represent the *initial* condition of the air entering the *wetted* portion of the coil.

By Equation 11, $M = \frac{1204.8 \times 12}{90{,}000 \times 0.243} = 0.661$

By Equation 9, $D_1 = 85.2 - 68.1 = 17.1$ deg.

Referring to Table 3, for $M = 0.661$, $\frac{D_2}{D_1} = 0.5164$.

$$D_2 = 0.5164 \times 17.1 = 8.8 \text{ deg}$$

From part (a), $t'_2 = 63.2$ deg.

By Equation 10, $t_2 = 8.8 + 63.2 = 72$ deg.

Occasionally the desired final wet bulb temperature is known, and the area of surface required to cool air must be found. In such a case, the boundary dry bulb temperature should first be computed by Equation 13a. If this computation shows that there will be dry surface, the area of the dry surface can then be computed by Equation 17, and the area of the wetted surface by Equation 1. The total surface required is the *sum* of the dry and wet surface areas. After the area of the wetted surface has been computed, the final dry bulb temperature of the air leaving the wetted portion of the coil can be computed.

The effect of the dry portion of the coil is exactly the same as though a separate precooling coil had been installed to cool the air down to the boundary dry bulb temperature, and a second coil installed for dehumidification. However, the two coils can be combined into one. There is no need for separate coils unless a different cooling medium, such as city water, is to be used only for precooling the air down to the boundary dry bulb temperature.

A coil part of whose area is dry will remove more heat than when its entire area is wet, although sensible heat

a = a numerical coefficient. See Table 1.
A_D = external area of dry surface, sq ft.
A_T = total external area of coil surface, sq ft.
A_W = external area of wetted surface, sq ft.
B = ratio of external to internal surface area of coil.
c = humid specific heat of air (average $c = 0.243$)
c' = specific heat of refrigerant.
d_L = large enthalpy (total heat) difference, Btu.
d_M = mean enthalpy (total heat) difference, Btu.
d_S = small enthalpy (total heat) difference, Btu.
D = wet bulb depression of the air, deg Fahr.
$D'' = t''_1 - t_{R_1}$, or $t''_2 - t_{R_2}$
$D_D = t_B - t_R$, or $t_D - t_{R_2}$
$D_D = t_1 - t_{R_2}$
D_L = large temperature difference, deg Fahr.
D_M = mean temperature difference, deg Fahr.
D_S = small temperature difference, deg Fahr.
$F = d_M/d_S$, or d_M/d_L. See text.
f_a = coefficient of heat transfer through air film, Btu per hr per sq ft per degree of temperature difference.
f_R = coefficient of heat transfer through refrigerant film, Btu per hr per sq ft per degree of temperature difference.
G = weight of air, lb per hr.
G' = weight of refrigerant, lb per hr.
h = enthalpy (total heat) of humid air, Btu per lb of dry air.
h_D = enthalpy (total heat) of air at boundary wet bulb temperature, Btu per lb of dry air.
h_R = enthalpy (total heat) of saturated air at temperature equal to the refrigerant temperature, Btu per lb of dry air.
h_{RB} = enthalpy (total heat) of saturated air at a temperature equal to the temperature of the refrigerant at the boundary, Btu per lb of dry air.
h_S = enthalpy (total heat) of saturated air at a temperature equal to the surface temperature, Btu per lb of dry air.
H_D = total heat transferred through dry surface of coil, Btu per hr.
H_W = total heat lost by the air flowing past the wetted surface, Btu per hr.
M = a numerical factor. See Equations 7, 11, and 24.
$N = \dfrac{Gc}{G'c'}$
$R = \dfrac{Bf_a}{f_R}$
t = dry bulb temperature, deg Fahr.
t' = wet bulb temperature, deg Fahr.
t'' = dew point temperature, deg Fahr.
t_D = boundary dry bulb temperature, deg Fahr.
t'_D = boundary wet bulb temperature, deg Fahr.
t_R = refrigerant temperature, deg Fahr.
t_{RB} = temperature of the refrigerant at the boundary, deg Fahr.
t_S = surface temperature of coil, deg Fahr.
U_D = over-all coefficient of heat transfer through dry surface of coil, Btu per hr per sq ft per Btu of temperature difference.
U_W = over-all coefficient of heat transfer through wetted surface of coil, Btu per hr per sq ft per Btu of enthalpy difference.
θ = angle in Fig. 4.

The subscript "1" refers to the initial condition of the air or refrigerant entering the coil. The subscript "2" refers to the final condition of the air or refrigerant leaving the coil.

will form a larger proportion of the total heat removed. Therefore, if the exact final dry and wet bulb temperatures need not be known, the performance of the coil can be computed as though the entire surface were wet—the result will be on the safe side insofar as the removal of total heat is concerned.

Area of Wetted Surface Required for a Counterflow Coil

As shown in Appendix 8, Equation 1 can be used to determine the area required for counterflow coils as well as for direct expansion coils.

The solution of counterflow problems is divided into three classes depending upon the value of the ratio $(Ga/G'c')$. When this ratio is *less than* one, the curves representing the enthalpy of the air and refrigerant (that is, the enthalpy of saturated air at the refrigerant temperature) at any point in their travel over the heat transfer surface are as shown in Fig. 9. The final wet bulb temperature of the air tends to approach the initial temperature of the refrigerant. This is by far the commonest case insofar as comfort air conditioning is concerned.

When $(Ga/G'c')$ is *equal* to one, the curves become two parallel straight lines as shown in Fig. 10. The enthalpy difference is the same at all points throughout the coil. The initial, final, and mean enthalpy differences are all equal to each other.

When $(Ga/G'c')$ is *greater than* one, the curves representing the enthalpy of the two fluids are as shown in Fig. 11. In this case, the final temperature of the refrigerant tends to approach the initial wet bulb temperature of the air.

In using Equation 1, the value of U_W is computed by means of Equation 3. As stated in an early part of this article, the value of a corresponding to the initial temperature of the refrigerant is used in Equation 3. The value of d_M, the mean temperature difference, is computed in the same way as illustrated in Example 2 except that the values of d_L and d_S are computed by the following formulas:

As illustrated in Fig. 9, if $(Ga/G'c')$ is less than one,

$$d_L = h_1 - h_{R_2} \quad \text{.....[18]}$$
$$d_S = h_2 - h_{R_1} \quad \text{.....[19]}$$

As illustrated in Fig. 10, if $(Ga/G'c')$ is equal to one,

$$d_L = d_S = d_M \quad \text{.....[20]}$$

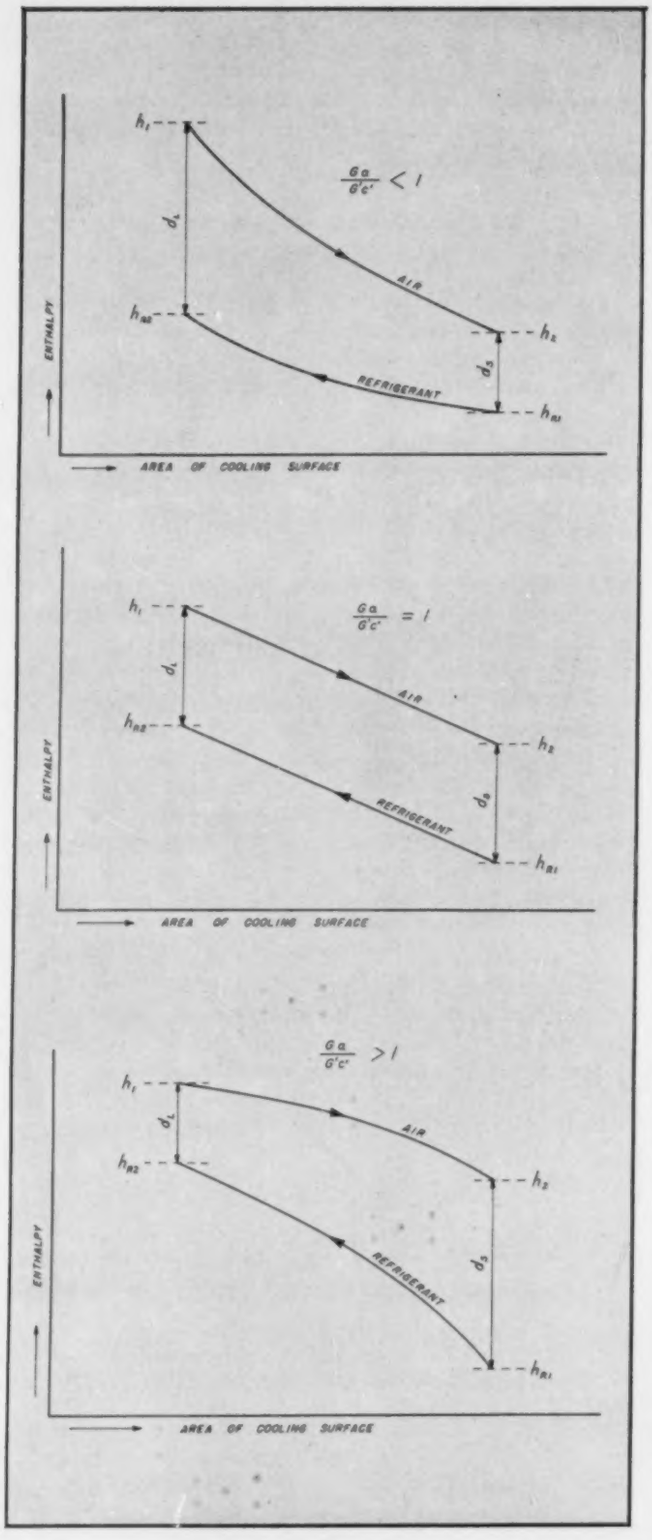
As illustrated in Fig. 11, if $(Ga/G'c')$ is greater than one,

$$d_L = h_1 - h_{R_2} \quad \text{.....[21]}$$
$$d_S = h_2 - h_{R_1} \quad \text{.....[22]}$$

Once d_M and U_W have been computed as outlined in the preceding paragraphs, the area of surface required is found by means of Equation 1 in exactly the same manner as illustrated by Example 3.

Final Wet Bulb Temperature of Air Leaving Wetted Surface of Counterflow Coil

A problem that frequently occurs is that of finding the total heat that a given coil can remove when only the initial wet bulb temperature of the air and the initial temperature of the refrigerant are known. The weights



TOP TO BOTTOM

Fig. 9—Diagrammatic illustration of change in enthalpy and temperature of air and refrigerant flowing through a counterflow coil when $(Ga/G'c') < 1$

Fig. 10—Diagrammatic illustration of change in enthalpy and temperature of air and refrigerant flowing through a counterflow coil when $(Ga/G'c') = 1$

Fig. 11—Diagrammatic illustration of change in enthalpy and temperature of air and refrigerant flowing through a counterflow coil when $(Ga/G'c') > 1$

of air and refrigerant flowing through the coil must, of course, also be known. When this is the case, the total heat transferred from the air to the water must first be computed by Equation 23. Once the total heat removed from the air is known, the final wet bulb temperature and the final temperature of the refrigerant can be computed. The derivation of Equation 23* is given in Appendix 9.

$$H_W = \frac{G\,(h_1 - h_{R1})}{1 + \dfrac{G}{A_W U_W F}} \qquad \text{.....[23]}$$

The interpretation of the symbol F in the above equation depends upon the value of $(Ga/G'c')$.

If $(Ga/G'c')$ is smaller than one, $F = d_M/d_S$.

If $(Ga/G'c')$ is equal to one, $F = 1$.

If $(Ga/G'c')$ is larger than one, $F = d_M/d_L$.

The values of $d/_M d_S$ and d_M/d_L can be found in either Table 2 or 3 after first computing the value of M by means of the following equation derived in Appendix 8:

$$M = \frac{A_W U_W}{G}\left(1 - \frac{Ga}{G'c'}\right) \qquad \text{.....[24]}$$

When $(Ga/G'c')$ is greater than one, the value of M will be negative. The negative sign should be disregarded when using Table 3. A negative value of M means only that $F = d_M/d_L$, whereas a positive value means that $F = d_M/d_S$.

Example 10: Air at an initial wet bulb temperature of 75 deg is to be cooled and dehumidified by a coil whose area is 1500 sq ft. The quantity of air to be cooled is 10,000 cfm. The quantity of water to be used is 150 gpm and its initial temperature is 48 deg. For this coil, $f_s = 10$, $f_R = 400$, and $B = 15$. Find: (a) the total heat transferred from the air to the water. (b) the final wet bulb temperature of the air. (c) the final temperature of the water.

Solution:

(a) First, the value of U_W must be found. Referring to Table 1, for $t_R = 48$ deg, $a = 0.593$. Using Equation 3,

$$\frac{1}{U_W} = \frac{0.243}{10} + 15 \times \frac{0.593}{400} = 0.04654$$

$U_W = 21.5$ Btu per Btu of enthalpy difference per hr per sq ft of surface.

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*This equation is analogous to Equation 6 on page 166 of the March, 1937, HEATING, PIPING AND AIR CONDITIONING. However, there is a slight difference in the interpretation of the value of F. In order to make Equation 6 of the March, 1937, article strictly analogous to the formula above, it can be derived in the following form in the same manner as Equation 23 above is derived.

$$H = \frac{Gc\,(t_1 - t_{R1})}{1 + \dfrac{Gc}{AUF}}$$

where t_1 = initial temperature of warm fluid;
t_{R1} = initial temperature of cold fluid;
G = weight of warm fluid;
c = specific heat of warm fluid.

The interpretation of F depends on the value of $(Gc/G'c')$.

If $(Gc/G'c')$ is less than one, $F = D_M/D_S$.

If $(Gc/G'c')$ is equal to one, $F = 1$.

If $(Gc/G'c')$ is greater than one, $F = d_M/d_L$.

This formula is to be used only for dry surfaces under the conditions discussed in the March, 1937, article.

Equation 23 is used to solve this problem. However, the value of M must first be found by means of Equation 24.

$$G = 4.5 \times 10{,}000 = 45{,}000 \text{ lb of air per hr}$$
$$(60 \times 0.075 = 4.5)$$
$$G' = 500 \times 150 = 75{,}000 \text{ lb of water per hr}$$
$$(8.33 \text{ lb per gal} \times 60 \text{ min} = 500)$$
$$\frac{Ga}{G'c'} = \frac{45{,}000 \times 0.593}{75{,}000 \times 1} = 0.356$$

Using Equation 24,

$$M = \frac{1500 \times 21.5}{45{,}000}(1 - 0.356) = 0.462$$

Inasmuch as $(Ga/G'c')$ is less than one, $F = d_M/d_B$. Referring to Table 3 for $M = 0.462$, the value of d_M/d_B is 1.271. Referring to Table 4 for $t'_1 = 75$ deg, $h_1 = 38.53$ Btu, and for $t_{B1} = 48$ deg, $h_{B1} = 19.18$ Btu. Equation 23 can now be used to find the heat removed from the air.

$$H_W = \frac{45{,}000\ (38.53 - 19.18)}{1 + \dfrac{45{,}000}{1500 \times 21.5 \times 1.271}}$$

$= 415{,}100$ Btu per hr total heat that can be removed by the coil from the air flowing through it.

(b) The heat removed from the air is

$$H_W = G\ (h_1 - h_2)$$
$$h_2 = h_1 - \frac{H_W}{G}$$
$$= 38.53 - \frac{415{,}100}{45{,}000} = 29.3 \text{ Btu, enthalpy of air leaving coil.}$$

Referring to Table 4, for $h_2 = 29.30$ Btu, $t'_2 = 64.1$ deg.

(c) The water in rising from its initial temperature of 48 deg to the final temperature t_2 absorbs 415,100 Btu per hr. The heat transferred to the water is

$$H_W = G'c'\ (t_{B2} - t_{B1})$$
$$t_{B2} = t_{B1} + \frac{H_W}{G'c'}$$
$$= 48 + \frac{415{,}100}{75{,}000} = 53.5 \text{ deg. final temperature of the water.}$$

Final Dry Bulb Temperature of Air Leaving a Counterflow Coil

The method of finding the final dry bulb temperature of the air leaving a counterflow coil is exactly the same as the method used for direct expansion coils. As shown in Appendix 10, Equation 8 is also used for computing the final dry bulb temperature of the air leaving a counterflow coil.

Example 11: Take the conditions of Example 10 and find the final dry bulb temperature of the air leaving the coil if the initial dry bulb temperature of the air entering the coil is 95 deg.

Solution: From Example 10, $t'_1 = 75$ deg and $t'_2 = 64.1$ deg. Using Equation 9, $D_1 = 95 - 75 = 20$ deg.

Using Equation 11, $M = \dfrac{1500 \times 10}{45{,}000 \times 0.243} = 1.37.$

Referring to Table 3 for $M = 1.37$, $\dfrac{D_2}{D_1} = 0.254.$

$$D_2 = 0.254 \times 20 = 5.1$$

Using Equation 10,

$t_2 = 64.1 + 5.1 = 69.2$ deg, final dry bulb temperature of air leaving the counterflow coil.

Boundary Dry Bulb Temperature for Counterflow Coils

As in the case of direct expansion coils, an initial portion of a counterflow coil may be dry. If a small quantity of water or brine is used, its temperature may rise to such a point that it is unable to maintain the coil surface at a temperature below the initial dew point temperature of the air. When this occurs, a portion of the coil will be dry and the condition curve will be similar to the one in Fig. 6. The following equations derived in Appendix 11 can be used to compute the boundary dry bulb temperature of the air.

$$D_B = \frac{(1 + R)\ D'' + ND_D}{R + N} \quad \text{.....[25]}$$

where:

$$D_B = t_B - t_{B2} \quad \text{.....[26]}$$
$$D_D = t_1 - t_{B2} \quad \text{.....[27]}$$
$$D'' = t''_1 - t_{B2} \quad \text{.....[28]}$$
$$N = \frac{Gc}{G'c'} \quad \text{.....[29]}$$

and as heretofore, $R = \dfrac{Bf_S}{f_B}$.

Example 12: Take the conditions of Examples 10 and 11 and find the boundary dry bulb temperature.

Solution: From a psychrometric chart, for $t_1 = 95$ deg and $t'_1 = 75$ deg, $t''_1 = 66.8$ deg. Using the data of Examples 10 and 11, $t_{B2} = 53.5$ deg, and

using Equation 27, $D_D = 95 - 53.5 = 41.5$ deg;

using Equation 28, $D'' = 66.8 - 53.5 = 13.3$ deg;

using Equation 14, $R = \dfrac{15 \times 10}{400} = 0.375$;

using Equation 29, $N = \dfrac{45{,}000 \times 0.243}{75{,}000} = 0.146$;

using Equation 25, $D_B = \dfrac{1.375 \times 13.3 + 0.146 \times 41.5}{0.375 + 0.146} = 46.7$;

using Equation 26, $t_B = 46.7 + 53.5 = 100.2$ deg.

Inasmuch as $t_1 = 95$ deg, the initial dry bulb temperature of the air is lower than the boundary dry bulb temperature. Hence, there will be no dry surface in the coil of Examples 10 and 11.

Notice that the boundary dry bulb temperature of the air cannot be found unless the final water temperature is known. This means in turn that the final wet bulb temperature of the air must be known. Hence, to find the condition of the air leaving a coil of *given area*, several final wet bulb temperatures must first be assumed. The final water temperature for each assumed final wet bulb temperature can then be computed. Next the boundary dry bulb can be computed for each of these different assumed final wet bulb temperatures. The area of dry surface can then be computed by Equation 17 for each assumed final wet bulb temperature. After this has been done, the area of wetted surface required to produce the various assumed final wet bulb temperatures can be figured by means of Equation 1. The areas of the dry and wet surface are then added to find the total surface required. Finally, the total surface required in each case is plotted against the final wet bulb temperature of the air. From such a curve the final wet bulb temperature of the air leaving coils of given area can be read.

In order to compute the area of dry surface in a counterflow coil by means of Equation 17, the value of t_{BB},

the temperature of the refrigerant at the boundary line, must be known (see Fig. 12). The value of t_{RB} can be computed by means of the following equation which expresses the fact that the heat surrendered by the air flowing past the dry surface is equal to the heat added to the water flowing through the dry portion of the coil.

$$Gc\ (t_1 - t_B) = G'c'\ (t_{R2} - t_{RB}) \quad \ldots\ldots\ldots\ldots\ldots [30]$$

The value of t_{R2} is found by first computing the total heat lost by the air in cooling from the initial to the assumed final wet bulb temperature. The value of t_{R2} can then be computed because all of the heat lost by the air is transferred to the water.

In computing the final dry bulb temperature of the air leaving the wetted portion of the coil with dry surface, the initial condition of the air entering the wetted portion of the coil is, of course, the condition of the air at the boundary.

When dry surface occurs on a counterflow coil and the value of both $(Gc/G'c')$ and $(Ga/G'c')$ is *less* than one, the curves representing the enthalpy of the air and water will appear as in Fig. 12. However, if $(Ga/G'c')$ is *greater* than one and $(Gc/G'c')$ is *less* than one, these curves will appear as in Fig. 13.

A set of refrigerant temperature curves on the psychrometric chart, similar to those of Fig. 7, is also of value in determining whether or not there will be dry surface on a counterflow coil. In this case the curves of refrigerant temperature represent the *final* temperature of the water or brine as shown in Fig. 14. The points for these curves are also computed by using Equation 13, except that for counterflow coils the values of D_B and of D'' are found by means of Equations 26 and 28 using the final refrigerant temperature.

Equation 13 can be used in this way to determine the highest initial dry bulb temperature that the air may have if the boundary line is to occur at the very edge of the air inlet to the coil—that is, wetting of the coil just commences at the point where the air enters the coil and the refrigerant leaves.

If the initial condition of the air is represented by a point to the left of the curve for the given *final* temperature of the refrigerant, there will be no dry surface. On the other hand, if the initial condition is represented by a point to the right of the *final* temperature curve, part of the surface will be dry. Although curves like those of Fig. 14 can be used to determine whether or not there will be dry surface, they cannot be used to determine the boundary dry bulb temperature of a *counterflow* coil. The temperature of the water increases as it flows through the coil; hence the temperature of the water at the boundary is lower than at the outlet of the coil. For this reason, the boundary dry bulb can be computed only by means of Equation 25.

Once the boundary dry bulb temperature has been computed by [25], the temperature of the refrigerant at the boundary can be read from curves like those of Fig. 14, instead of computing it by Equation 30. For example, if the final temperature of the refrigerant is 50 deg. and the initial condition of the air is represented by point "1" of Fig. 14, part of the surface will be dry. If the refrigerant temperature were constant at 50 deg as in a direct expansion coil, the boundary condition would be represented by point "2." However, inasmuch as the refrigerant temperature is changing, the boundary dry bulb is computed by means of Equation 25 and located on Fig. 14 at point *B*. The temperature of the refrigerant at the boundary, that is the value of t_{RB}, can then be read from the curve of refrigerant temperature passing through point *B*.

A set of curves like those of Fig. 14 is also useful for determining the highest permissible water temperature if the possibility of dry surface is to be avoided on a counterflow coil. For example, if dry surface is to be avoided when the initial condition of the air is represented by point "1," the highest permissible *final* water temperature would be read from the curve passing through point "1."

Surface Temperatures in Counterflow Coils

As in the case of direct expansion coils, the condition curve for a counterflow coil can be drawn either by plot-

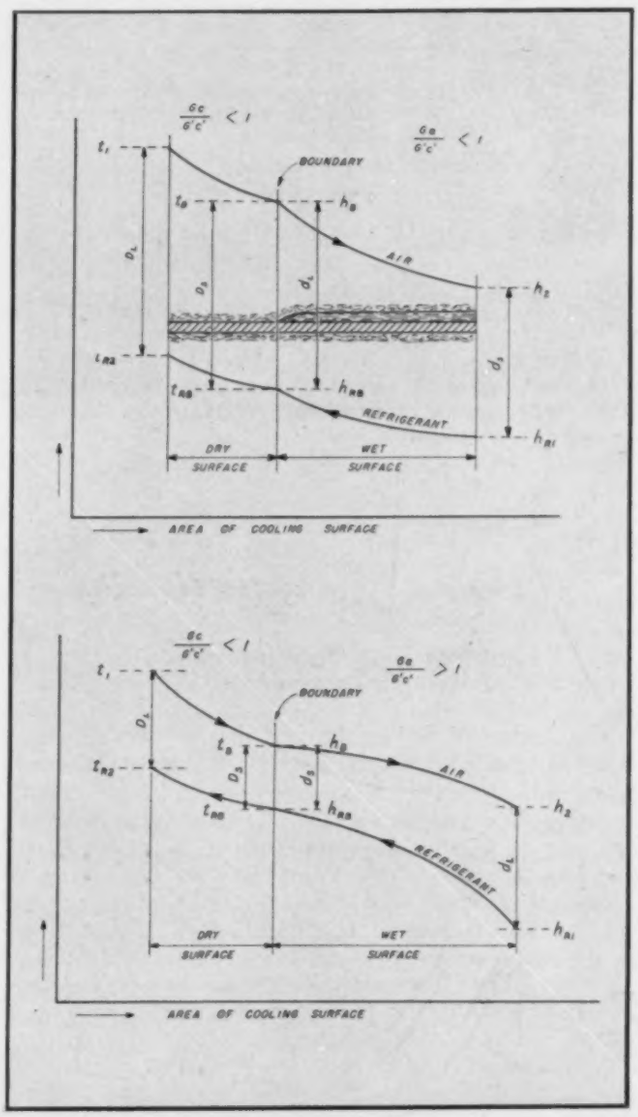


Fig. 12 (above)—Diagrammatic illustration showing change in the temperature and enthalpy of the air as it flows over a counterflow coil that is partly dry. For this case $(Gc/G'c')$ and $(Ga/G'c')$ are both less than one

Fig. 13 (below)—Diagrammatic illustration showing change in the temperature and enthalpy of the air as it flows over a counterflow coil that is partly dry. For this case $(Gc/G'c')$ is less than one whereas $(Ga/G'c')$ is greater than one

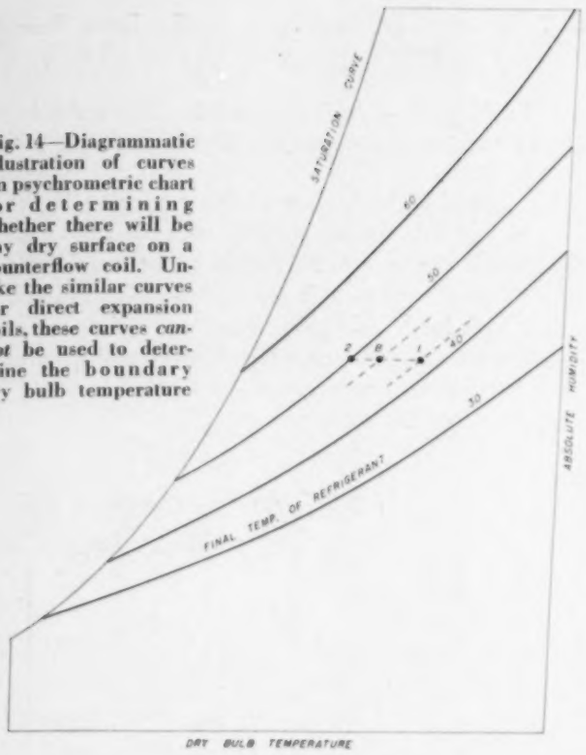


Fig. 14—Diagrammatic illustration of curves on psychrometric chart for determining whether there will be any dry surface on a counterflow coil. Unlike the similar curves for direct expansion coils, these curves *cannot* be used to determine the boundary dry bulb temperature

ting the computed final dry and wet bulb temperatures, or by determining the surface temperatures by means of Fig. 3.

In order to determine the surface temperatures of a counterflow coil for various wet bulb temperatures, the temperature of the refrigerant corresponding to each of the wet bulb temperatures must be known. Although they can be computed, the need for doing so can be avoided by drawing a line such as "1-2" of Fig. 15. Point "1" is located at the intersection of the initial wet bulb temperature of the air and the final refrigerant temperature. Point "2" is located at the intersection of the final wet bulb temperature of the air and the initial temperature of the refrigerant. Any point on the line "1-2" gives the temperature of the refrigerant for any wet bulb temperature. To find the surface temperature corresponding to the wet bulb temperature through point "3," draw the line "3-4" at an angle θ, which is found in Table 5 after first computing (cf_R/Bf_E) for the counterflow coil. Then read the corresponding surface temperature at point "4." By using the line "1-2," there is no need even to read the refrigerant temperature corresponding to each wet bulb. This operation can be repeated at any number of points along line "1-2," thus determining the surface temperatures corresponding to different wet bulb temperatures of the air. To find the *highest* surface temperature on the coil, draw the slanting line of Fig. 15 through point "1" which represents the highest wet bulb and refrigerant temperatures of the coil.

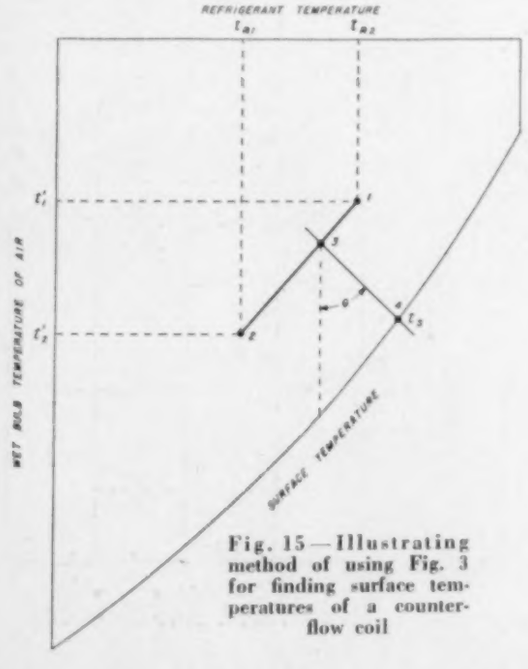


Fig. 15—Illustrating method of using Fig. 3 for finding surface temperatures of a counterflow coil

After the surface temperatures have been found as described in the preceding paragraph, the condition curve can be drawn for a counterflow coil in exactly the same manner as previously described for a direct expansion coil. The condition curve obtained by this method will agree closely with the curve obtained by plotting the computed final dry and wet bulb temperatures.

Because the values of a in Table 1 were computed on the assumption that the difference between the highest surface temperature and the initial refrigerant temperature will be less than approximately 20 deg. the temperature rise of the refrigerant must also be less than 20 deg. Practically, this is no limitation because the temperature rise of the refrigerant while travelling through the wetted portion of a counterflow coil is very rarely as large as 20 deg. A considerable part of the counterflow coil will usually be dry if the temperature rise of the refrigerant is as large as this. Therefore, even though the temperature rise of the refrigerant while travelling through the entire coil may be more than 20 deg. the rise while travelling through only the wetted part is usually considerably smaller.

[To be continued]

Flour Mill Air Conditioning—

[Concluded from page 5]

Hills region it is probable that snow will be on the ground most of the winter when the sprays are not in operation. Thus the cleanliness of the outside air entering the mill is assured as closely as possible under the circumstances.

The main fan delivers air to a duct distributing system at the ceiling of the roll floor. Experience has shown that the demand for cooling in summer is centered on the roll floor, and hence this and part of the second floor are the only parts of the mills served directly by supply ducts. Conditioned air, warmed up a little, of course, has free access to other parts of the mill through gratings in the floors. This air is drawn up to the purifiers by the steady suction of the purifier fans; part of it drops through to the packing floor by gravity, and in general diffuses fairly evenly throughout the operating part of the mill. Fifty per cent of the total air handled again enters that part of the milling machinery having suction apparatus; the other 50 per cent may go back to the air conditioning room through the return opening, and the cycle of the air is complete.

Area and Weight of Ducts . . . . . By Peter Franck*

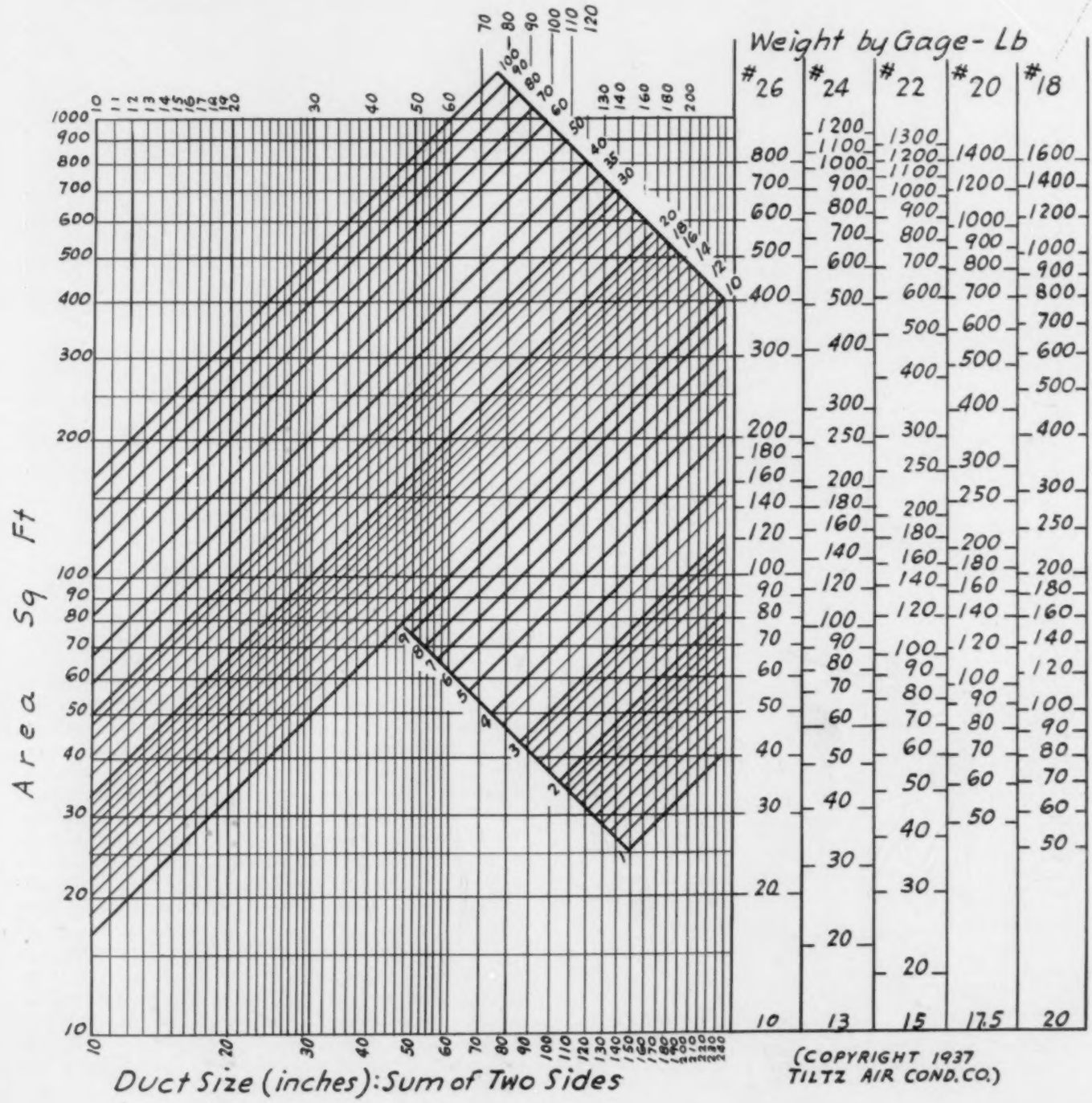
THE graph reproduced here was prepared to facilitate the determination of sheet metal duct surfaces and weights. From it may be read the weights of the various gages and the area for any length of run of a rectangular duct.

The bottom scale represents the sum of the two sides of the duct and the oblique lines represent the length of run in feet. Going horizontally to the right from the intersection of the vertical and oblique lines, the weight for No. 26 gage is read on the first scale. The square feet of surface may also be read on this scale, since—for No. 26 gage—1 sq ft weighs 1 lb. The scales farther to the right give the weights of the duct run for the heavier gages.

By following the horizontal line corresponding to an area of 1 sq ft to its intersections with the various scales for the different gages, the weight of the metal per sq ft may be read; such as 1 lb per sq ft for No. 26 gage, 1.3 lb per sq ft for No. 24 gage, etc.

Since all the scales are logarithmic, multiples of 10 may be used on any scale, provided this is taken into account when the answer is read.

*Tiltz Air Conditioning Corp.



Welding Carbon-Moly Pipe at Delray

Preventing Air Hardening . . . Need for Preheating . . . Preheating Procedure . . . Stress Relief

By **Sabin Crocker***

But limited information has been published on the welding of carbon-molybdenum pipe — a material which possesses several advantages which make it popular for use in the temperature range of 850 to 950 F. Carbon-moly steel is being used for the 865 lb, 910 F steam piping in the extension of the Delray Station; consequently, it is possible to report on the Company's experience in welding this piping material to supplement available data. . . . Advantage of the opportunity is taken in this and succeeding articles to describe also new welding practices and construction details adopted during the course of this piping project

CARBON - MOLYBDENUM steel is being used for the 865 lb, 910 F steam piping in the extension of the Delray Station (Fig. 1) of The Detroit Edison Co. described in HEATING, PIPING AND AIR CONDITIONING for January, 1938.[1] As mentioned there, this steel is of the ½ per cent molybdenum, 0.10 to 0.20 per cent carbon variety covered by ASTM Specification A206-38T which has been generally adopted for recent central station and industrial projects having superheat temperatures of 850 to 950 F. Carbon-moly steel possesses three advantages which account for its popularity in this temperature range: (a) creep strength at 850 to 950 F comparable with plain carbon steel at 750 to 850 F; (b) among alloy steels a weldability approaching that of carbon steel; (c) relatively small alloying content with correspondingly low price for an alloy steel (approximately two to three times that of plain carbon steel on a per pound basis). Unfortunately, however, little if any advantage can be claimed for carbon-moly steel in most media at temperatures up to 950 F, on the score of resistance to corrosion, which is reported to about equal that of carbon steel.

Fig. 1—Delray plant during construction of extension

Although Detroit Edison's brief experience in the use of carbon-molybdenum pipe is not wholly novel, nevertheless an account of what has been done at Delray will supplement and amplify the limited information which has been published to date about welding this new pipe material. At the same time advantage will be taken of the opportunity to describe new welding practices and construction details adopted during the course of this job. A section of the carbon-moly main steam header of the Delray extension is shown in Fig. 2 as it appeared before heat insulation was applied.

Past Welding Practice

In Detroit Edison's Conners Creek rebuilding project, which immediately preceded the Delray extension, silicon killed, medium carbon steel pipe purchased to ASTM Specification A106, with supplementary requirements on number of tests, was found ample for steam conditions of 650 lb and 850 F. Here was the Company's initial use of welded joints for high pressure piping, with the development of a comprehensive practice in welding carbon steels which provided a background of experience for extending the technique to suit alloy piping at Delray. As set forth in previous articles,[2] the salient features of Conners Creek welding were: qualifying operators through rigorous tests; using the arcronograph for quality control; depositing metal from coated electrodes in numerous layers, each carefully cleaned and chipped; stress relieving with an electric induction heater affording close temperature control; taking advantage of the possibilities of welded fabrication through originating novel details not feasible with older types of construction. While valves of nickel-chromium-molybdenum steel and carbon-molybdenum steel were welded into the lines, the attachment was made with a mild steel electrode to plain carbon steel pipe and there were few, if any, instances where one alloy steel part was directly joined to another by welding. This favorable situation alleviated any tendency for the alloy constituents to cause trouble through air hardening in welding, since metal from the rod tended to dilute the alloy content of the fusion zone to a point where air hardening was negligible.

Preventing Air Hardening

At Delray, however, where it became necessary to weld pipe-to-pipe joints of carbon-moly and pipe-to-valve joints of carbon-moly to nickel-chrome-moly steel with a rod having molybdenum in its coating, some thought had to be given to adopting means for avoiding air hardening of the fusion zone during cooling. To begin with, of course, it was necessary to train and qualify operators in the use of carbon-moly electrodes for welding alloy members of the sort indicated. Fortunately this was no

*Senior Engineer, Engineering Division, The Detroit Edison Co. Member of Board of Consulting and Contributing Editors.

[1] "Modern Practice in Piping Design, the Steam and Boiler Feed Piping for the 865 Lb, 910F Extension of Delray," by Sabin Crocker, HEATING, PIPING AND AIR CONDITIONING, January, 1938.

[2a] "An Examination of Welds Made Under Field Conditions for High Pressure, High Temperature Steam Station Piping," by A. E. White, D. H. Corey and C. L. Clark, *Journal*, American Welding Society, September, 1934.

[2b] "Welding High Pressure, High Temperature Steam Piping," by Sabin Crocker, HEATING, PIPING AND AIR CONDITIONING, February and March, 1935.

[2c] "Welded Fabrication of Manifolds and Special Fittings for Piping," by Sabin Crocker, HEATING, PIPING AND AIR CONDITIONING, January, 1937.

[2d] "Modernizing the Conners Creek Power Plant," by Sabin Crocker, *Combustion*, March and April, 1936.

[2e] "Arc Welded Piping in Central Station Steam Plants," by C. F. Hirshfeld and D. H. Corey, *Advance Paper*, Society of Naval Architects and Marine Engineers, 1936.

[2f] "Grading Welders," by D. H. Corey, *Power*, July, 1938.

great problem since competent personnel were available who quickly adapted themselves to the slight change in application required. Fig. 3 shows an operator welding carbon-moly pipe to a nickel-chrome-molybdenum valve with an electric resistance preheater in use.

The factors tending to produce air hardening or brittleness in a weld are, first, the presence of a sufficient amount of carbon or alloying element to impart hardenability to the steel; and second, too rapid a rate of solidification and cooling. The measures taken to combat excessive hardening in welding a low alloy steel such as carbon-moly are: (a) for a given alloy composition keep the carbon content as low as is compatible with obtaining adequate strength, preferably below 0.20 per cent and in all cases below 0.35 per cent; (b) preheat the parts being joined in and adjacent to the fusion zone in order to delay cooling through the critical region, thus obtaining decreased hardness with greater ductility.

Fig. 2—Carbon-moly main steam header before heat insulation was applied. For line diagram of steam piping see Fig. 3 of Footnote 1

Need for Preheating

Some question exists as to the need for preheating a ½ per cent molybdenum steel where the carbon content is below 0.20 per cent, as is the case with pipe bought to ASTM Specification A206. The carbon content of carbon-moly or nickel-chrome-moly cast or forged steel valve bodies or welding flanges, however, is of the order of 0.25 to 0.35 per cent, which calls for greater care in the case of welds involving these members. The committee report on "Low Alloy Steels" given in Chapter 25 of the American Welding Society's "Welding Handbook," 1938, p. 492 has the following to say on the weldability of ½ per cent molybdenum steel:

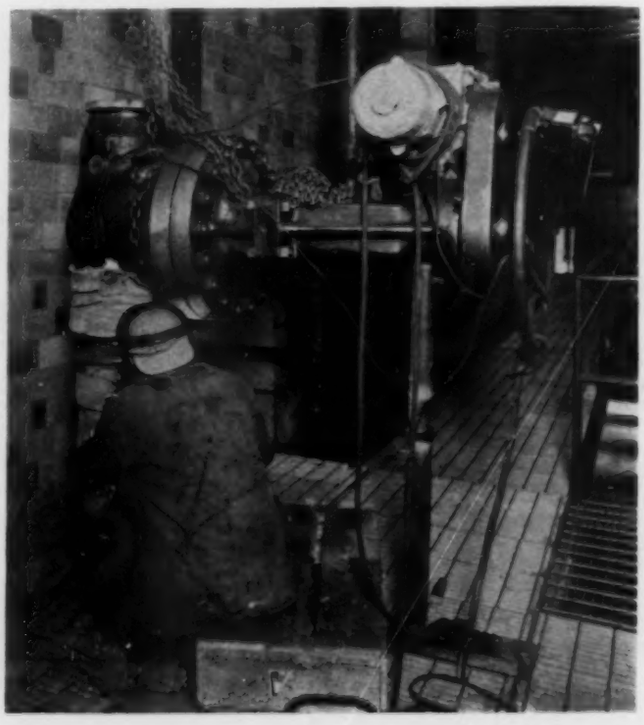
> The ½ per cent molybdenum steels have excellent weldability provided the carbon content is low. No serious difficulties have been encountered in welding even heavy plate structures of this steel. A number of welded pressure vessels for oil refinery service and for superheater drums have been constructed to U-68 requirements (ASME Boiler Code). The steel has also been used for a considerable amount of welded power plant piping for the steam lines from the boiler to the turbines. These lines generally involve cast valves of ½ per cent molybdenum steel but with carbon content as high as 0.30 per cent. This higher carbon content in the ½ per cent molybdenum valves necessitates preheating in welding to the valve. Preheating for carbon contents in excess of 0.15 per cent is generally recommended, the higher the carbon content the greater the preheating temperature employed.

Little actual evidence exists to show what tangible benefits if any are derived from preheating carbon-moly steel having a carbon content less than 0.20 per cent with molybdenum around ½ per cent. Microscopic examination of welds in 0.15 per cent carbon, 0.50 per cent molybdenum pipe made with and without preheating showed little superiority for welds in preheated pipe.[3] The grain structure of the welds made without preheating was termed *sorbitic* while the welds which were preheated were classed as *pearlitic*. Both structures were considered to be entirely acceptable. Hardness surveys of sections of single bead deposits on carbon-moly showed somewhat lower hardness values for the preheated specimen. Hardness traverses of a weld in 1⅛ in. wall carbon-moly pipe showed much higher values in the first bead deposited without preheat than in the first bead of the preheated weld, but the hardness of subsequent beads was not significantly different. Maximum hardness in the preheated weld occurred in the top bead and was equal to the hardness found in the bottom bead of the unpreheated weld.

Nevertheless, it is generally conceded that preheating is more essential with higher alloy steels, or where the carbon content is over 0.20 per cent in carbon-moly steel, or over 0.35 per cent in plain carbon steel. For instance The Detroit Edison Co. had an experience which definitely confirmed this opinion when the end of a small socket welding valve snapped off just back of the fusion zone while the final bead was being chipped and cleaned. Subsequent investigation disclosed that the valve was not cast carbon-moly steel as had been supposed, but that the manufacturer by mistake had supplied a cast chrome-manganese-moly body intended for flanged con-

[3]Review of literature on "Welding Molybdenum Steels," by W. Spraragen and G. E. Claussen, supplement to the *Journal*, American Welding Society, November, 1938, p. 15.

Fig. 3—Operator welding carbon-moly pipe to a 10 in. nickel-chrome-molybdenum valve with an electric resistance preheater in use



struction which had definite air hardening tendencies.

It is interesting to note that tests reported by R. W. Clark[4] indicate that preheating tends to obviate some of the locked in stresses which otherwise need to be relaxed through stress relief. From these tests Mr. Clark drew the inference that stress relief is not quite so necessary where preheat is employed, although some benefit is still derived therefrom.

Procedure in Preheating

As in the case of most new developments, considerable difference of opinion has arisen about what constitutes adequate procedure for preheating carbon-moly, as well as under what circumstances it needs to be preheated at all. The conservative tendency has been to play safe at the outset by taking more precautions than eventually may turn out to have been necessary. One moot question involves whether preheat is needed only while weld metal is being deposited, or whether temperature should be maintained from start to finish of the weld and sustained until stress relieving is accomplished. Advocates of the first-mentioned, and easier, school of thought are content to shut off preheat and discontinue work overnight or week ends in case a weld requires more than one working day, while some of those who favor the "hard way" even go so far as to require the operator or operators to weld continuously without shutdown until the joint is completed.

Not wishing to establish welding procedure more difficult than necessary, the Company's engineers made extensive inquiries as to the experience of others in preheating carbon-moly welds, seeking particularly to find out what evidence existed that a definite advantage resulted from preheating and/or welding continuously from start to finish of each joint. No definite proof was forthcoming that any actual benefit was derived from doing the job the hard way, while several convincing demonstrations were offered that preheat only during actual welding was sufficient, and that both welding and preheat could be interrupted without damage after the first four or five beads were laid in a joint. Accordingly the following procedure for preheating and stress relieving has been followed for the first unit of the Delray extension:

(a) The ends of the parts to be joined by butt welding are preheated uniformly to a temperature of 400 to 600 F with an electric resistance heater before welding is begun.

(b) A metal temperature of 400 to 600 F is maintained while weld metal is being deposited or successive layers cleaned and chipped, an electric resistance heater being employed so that heat can be applied at the same time electric arc welding is going on.

(c) In the case of large welds which cannot be finished in one shift, at least five layers are deposited before preheating is interrupted. After welding has progressed this far, preheat may be cut off during the night or over the week end provided the temperature is brought back to 400 to 600 F before resuming welding.

(d) Following completion, each butt weld is annealed as soon as convenient with an electric induction stress reliever, except that built up manifolds or large branch connection (fish mouth) welds made in the shop are taken out to a heat treatment furnace. Stress relieving of carbon-moly is carried on at a temperature of 1200 F, plus or minus 25 F, for a period of two hours per inch of thickness of the material.

(e) Fillet welds used with pipe sizes 2 in. and smaller are preheated and stress relieved with a fuel fired torch. This includes socket welded line joints as well as the attachment of small drip and instrument lines to large pipes or manifolds (see Fig. 2). Present practice is to fillet weld the nipples for bypass connections to alloy steel valve bodies before they are heat treated, thus effecting stress relief of the weld at the same time.

[4] "Welded Joints in Carbon-Molybdenum Piping," by R. W. Clark, HEATING, PIPING AND AIR CONDITIONING, November and December, 1938.

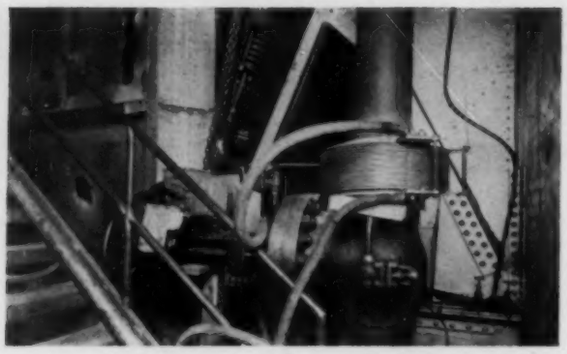
Resistance Preheater Used

Fig. 4 is a photograph of electric resistance preheating equipment used at Delray, with the asbestos blankets removed to show the heating elements. The resistance type is used for preheating in preference to an induction heater on account of the unfortunate disturbance of the welding arc set up by the magnetic field of the latter if and when the preheater is in use while metal is being deposited. Although it is true that some concerns have successfully employed induction heating for this purpose through shutting off the preheater while welding was under way and then turning it on again while successive beads were being cleaned, the results thus obtained fall short of what can be accomplished with a resistance heater. The chief objection is that residual heat plus what is developed by one arc in welding is not sufficient on large pipes to hold the temperature within bounds while a bead is being completed and it becomes necessary to have two welders operate simultaneously to secure the requisite heat input. Two operators working on the same joint means numerous interferences, plus the added complication of divided responsibility for the work.

Fig. 4—Electric resistance preheater in place on 10 in. pipe-to-valve joint, with asbestos blanket removed to show heating elements

Stress Relief with Induction Heater

The electric induction heater developed by Detroit Edison in 1933 for stress relieving welded pipe joints during rebuilding of the Conners Creek plant[2b] proved so satisfactory that its use on high pressure joints at Delray was a foregone conclusion. A composite photograph showing the essential parts of this device and its appearance in use on a 10 in. valve built to the American Standard for 900 lb SSP is shown in Fig. 5. The electrical circuit for the device is shown schematically in Fig. 6. In addition to the tap changing adjustments on the primary and secondary coils of the transformer, full automatic temperature regulation is obtained through a control circuit which actuates a magnetic contactor in response to the indications of a thermocouple attached to the weld.



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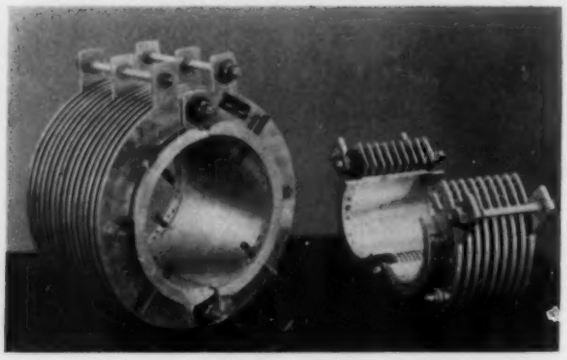

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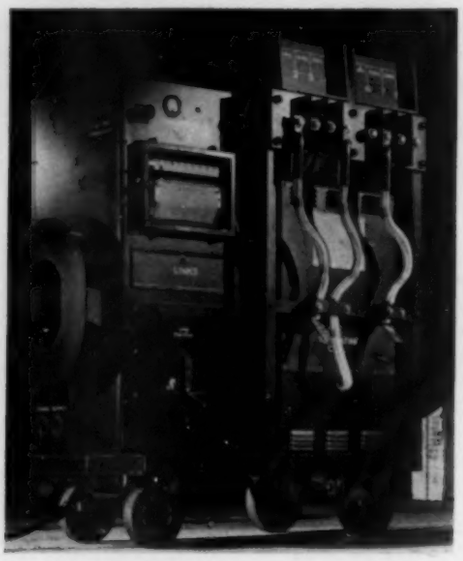


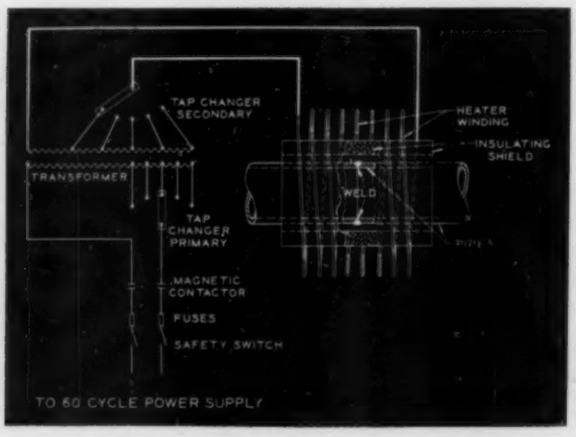
c

Fig. 5—Electric induction stress reliever. (a) Induction heating collar in place on 10 in. valve. (b) Collar removed for display. This collar is the primary winding of the induction heater in which the pipe joint serves as an electrical secondary short circuited on itself. (c) Portable mounting for power transformer and automatic control equipment

Company instructions call for stress relieving carbon-moly welds at a temperature of 1200 F, plus or minus 25 F, for a period of two hours per inch of wall thickness. This differs from the practice for preheating carbon steel at both Conners Creek and Delray where instructions call for holding a temperature of 1100 to 1200 F for one hour per inch of wall thickness. Slow cooling with the insulating jacket in place is considered desirable until the temperature falls below 600 F, when the jacket can be removed and the joint allowed to cool in still air. Holding full temperature throughout a band having an overall width about twice that of the weld is believed to be sufficient, since this embraces metal which may be benefited by the drawing temperature and which has the highest locked in stresses. Restricting the width of the band heated to full stress relief temperature is advantageous in that it avoids severe heating of the valve body in the case of pipe-to-valve joints, thus precluding warping the valve mechanism through excessive temperature. Typical temperature gradients across the heated band and minute to minute temperature variations at the control thermocouple are shown graphically in Fig. 7. The maximum point to point variation in temperature around welds in 12 to 16 in. pipe resulting from variations in surface radiation, etc., has been found by investigation with thermocouples to be within the ± 25 F leeway permitted by the instructions. The variation on smaller welds tends to be still less.

[A second article will appear next month]

Fig. 6—Schematic diagram for electrical circuit of induction stress reliever



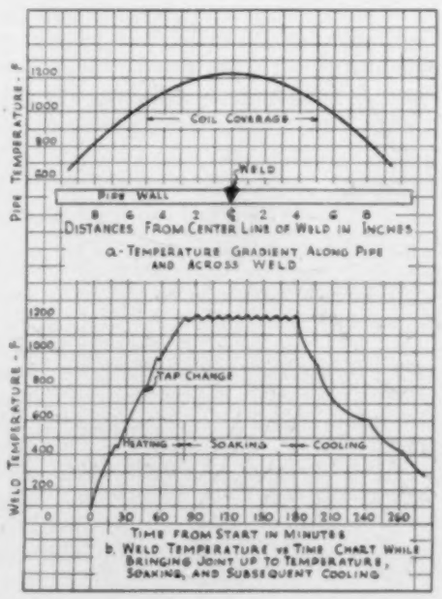

Pipe Temperature - F
Coil Coverage
Weld
Pipe Wall
Distances From Center Line of Weld in Inches
a- Temperature Gradient Along Pipe and Across Weld
Weld Temperature - F
Tap Change
Heating
Soaking
Cooling
Time from Start in Minutes
b. Weld Temperature vs Time Chart While Bringing Joint up to Temperature, Soaking, and Subsequent Cooling

Fig. 7—Typical temperature conditions observed in stress relieving a weld in 10 in. diameter 0.718 in. wall (Schedule 100) carbon-moly pipe. (a) Temperature gradient along pipe and across weld. (b) Weld temperature vs. time chart while bringing joint up to temperature, soaking, and subsequent cooling

Heating and Cooling Engineer Talks About Some Troubles

By Samuel R. Lewis*

Anyone can tell about the virtues of some particular plant, but Mr. Lewis describes here troubles with various installations and explains what he would do differently now in the light of these experiences. . . . His article is an unusually frank and informative discussion of many of the practical matters involved in the design, installation, operation and maintenance of heating and cooling systems

ANYONE can tell about the virtues of some particular plant. I was asked to describe a large heating and cooling system which was operated through one winter and one summer. It has made no serious complaint and has been paid for. However, it seems more interesting to me to tell about this plant in the light of what about it I would do differently were I to repeat the performance.

The plant has an exhaust fan for every supply fan and their functions are interlocked. In designing it, I knew enough to arrange for adequate zoning so that both in winter and in summer the heat input and output could follow the progress of the building against the sun. I knew enough to allow for control of heat input and output to compensate for variations in occupancy of the many different rooms and for variations in heat reception from manufacturing processes. I knew enough to arrange for use of less convector area in winter than in summer, due to the higher transfer rate because of greater temperature differences.

I did not know enough to fight for more clearances around the convector units so that doors, easily opened by latches and swinging on hinges, would permit easy access for maintenance. Instead, I designed a plant in which, to get at the convectors for cleaning or to change air filters, one must remove a lot of screws, build a scaffold, and lift down from overhead a heavily insulated steel plate bottom.

I failed to appreciate that in a big system which uses water the year around, a sweated copper joint in a large sized main, which leaks, cannot be repaired without draining the entire system, involving many hours and the loss of hundreds of gallons of water. It is true that a similar leak in screwed or welded iron or steel might have required equal drainage for an adequate repair, but there are some chances of caulking or banding such a leak, which chances are denied with copper. It is always wise to install many valves on such mains so that leaks can be isolated.

I learned, in another large installation, that a finned copper convector with its small horizontal tubes will not drain itself so dry but that enough water may be retained by capillary or some other action to freeze and burst the tube. We learned to blow out the water with high pressure compressed air, but even then one cannot be sure but that some water remains. The answer here seems to be to employ an antifreeze solution, and to keep the pump going. I do not say that the same retention of water and subsequent freezing might not have occurred with leaky outside air intake dampers and a steam heated convector, because I have had just this happen.

In another plant where water cooled fin convectors were installed, I thought I would be economical and smart if I employed the same convectors the year around, using cold well water in summer and recirculated heated water in winter. No trouble ensued in winter, but unfortunately in summer the anticipated occupancy and heat load from process eventually to be anticipated, was lacking in 1938. As a result, even though the bypass dampers, expected in summer to reheat and dilute the cool, damp air emerging from the convectors, were open wide, the rooms were too cold if we chilled the convectors enough to bring the air below its dew point temperature. A measure of success was achieved by reducing greatly the volume of air delivered, an unusual prescription in a cooling system, but were I to redesign that plant there would be a separate steam or hot water reheater between every cooling convector and every room served by that convector. (It may be that 1938 was unique in having rather high general outdoor relative humidity and many days of comparatively cool dry bulb temperature.)

Corrosion of Condensers

In another plant (which fortunately has none of the foregoing troubles) which has reheaters to leeward of its cooling coils, with cold water pretempering and direct expansion recoolers, with every conceivable human contingency well forefended, an entirely new difficulty has emerged. There is a deep well of 500 gpm capacity supplying water at 55 deg. The basically intelligent scheme is to send this water to meet the entering air and to cool it as far as possible, followed by the sharp dehumidifying cooling of the evaporators from a "Freon" refrigerating system.

It seemed clever to employ the water from the well, following its use in absorbing the first heat from the air, to condense the refrigerant gas in conventional finned copper self-contained condensers. The plant was installed in 1936 for about half of the large building and all was so satisfactory that the balance of the building was given complete air conditioning ready for 1938 summer service. This involved adding 100 tons more refrigeration capacity. After less than 60 days' operation the condensers of the new "Freon" compressor gave up the ghost, the entire refrigerant charge was lost, and the refrigerant system was flooded with water. This

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An address presented at the November, 1938, meeting of the Illinois Chapter of the American Society of Heating and Ventilating Engineers.

flooding, of course, was a major calamity, involving weeks of draining and evacuating to get rid of the moisture from inside the piping and convectors. The well water has proved to be thoroughly polluted with hydrogen sulphide.

Why the 1936 condensers, sworn duplicates of the 1938 condensers in material and thickness but which gave way so quickly, have not gone, is somewhat a mystery. There is disagreement among metallurgists as to whether any metal short of monel will withstand for long the combination of hydrogen sulphide water and an increase, however small, in temperature. Of course, the condenser water was never what one could call hot, and during more than half of the time the condenser which gave way was not exchanging any heat at all.

"Why the 1936 condensers have not gone is a mystery."

The saving in this case by using the large quantity of cool water is too great to abandon the use of the well. The answer may be to install an evaporative condenser, and to employ city water for making up evaporation—thus eliminating all contact between the refrigerant system and the well water, but continuing the use of the latter in the tempering convectors, where it accomplishes something like 200 tons of refrigeration. If these tempering coils eventually are destroyed they can be replaced, and the cost of replacement every few years will be justified. There will be no water in the refrigerating system even if they do leak. It may be that with the relatively mild heat present in this service the destruction of the metal will be very slow.

This experience recalls a practice said to have been adopted by one large chain store system which purchases scores of automatic condensing units; they refuse to accept the conventional welded-in, highly efficient—but decidedly inaccessible—condensers, and require in lieu thereof detached shell and tube condensers of much greater size and cost.

In the case I have described, since the owner had purchased new condensers before I was called in, we have decided in favor of a steel tube heat transfer device, somewhat after the order of a water heater, and will circulate the sweet condenser water through the tubes, with the sulphide water around them. Such a heater can be opened easily and inspected at the end of every cooling season and the occasional retubing will not be too costly. The chances of getting water in the refrigeration system will be greatly reduced. A big evaporative condenser, necessarily in this case placed on the roof and exposed to smoke and the elements, is itself not exactly a permanent investment.

Increased Occupancy of Store Basement

Another complete air conditioning plant had an adequate ventilating and cooling system serving a basement, first and second stories of a department store. Every human contingency that I could think of was taken into account. The differential control of the first and second stories was met by having about half of the heat removal on each cared for by local recirculating, cooling and dehumidifying machines. The basement, only about half of which was in use, received no solar heat and by all calculations should be satisfied with air at the same temperature and adjustable volume as that for the first story, coming from the central ventilating plant. This proved true for the first summer only, after which the owner, finding the basement so comfortable and attractive, proceeded to more than double the illumination, to increase greatly the basement selling floor area, and to attract many more people to the basement. As a result, when the entering air was cooled sufficiently to serve the basement, as was possible, the first story became too cool, even while the auxiliary recirculating units, intended to balance the presumably greater cooling demand of the first story, were shut down. It was not practicable to achieve temperature control by reducing and adjusting continually the volume of air going to the first story.

There were so many days and evenings during 1938 when no cooling whatever was needed for this store except in the basement, that we are now installing an entirely separate basement system. I could just about as well have done this in the beginning. I pray that I may never again fail to do so in a similar case.

Had I used the same common sense I really had at the time of the original design (because I did use it on contemporary plants) I would have made arrangements permitting delivery of air at any reasonable range warmer or cooler than the average, to any room or story or department regardless of the temperature of the air being sent at that time to other departments.

In another air conditioned store, happily not designed by myself, complaint was made to me that while they could get cool, the place smelled. Upon investigation I found indeed that it did smell. The plans indicated an adequate outside air intake and adequate convectors for cooling and dehumidifying a mixture of at least one-third outside air with two-thirds recirculated air. However, while the plans showed an adequate exhaust fan, it had never been installed.

Both side walls were party walls without any windows. The front end of the store had one double door with automatic closers and no windows. The back of the store had one door, necessarily kept locked, and no windows in communication with the air conditioned spaces. Even though the outside air intake dampers were opened wide, no measurable amount of new air would enter, because the building already was full of air which could not escape. Needless to say, we compelled the installation of the exhaust system.

Difficulties in Handling Air

Many years ago I installed an air washer (using refrigerated water) to cool a building which had an existing ventilating system of about 100,000 cfm capacity which was situated in the basement without any means of recirculating the air. The spent air in this old plant entered vent flues which opened into the attic above the fifth story, from which the air in winter escaped adequately through a number of roof ventilators. After installing the cooling apparatus, I visited the attic on a hot summer day and was surprised to observe that, whereas in winter a gale of smelly air escaped through the roof ventilators, no air was departing through them during the cooling season. Evidently the supply fan

was unable to force the much heavier cool air up and out of the building into the very hot expanded outdoor atmosphere, and I understood why, as each successive story above the first one was examined, the cooling was less and less effective.

We took steps, of course, as soon as possible to install exhaust fans in the attic and to build an adequate return air shaft from them back to the basement.

I have observed that those cooling plants which have the fans and cooling surfaces above the spaces which are to be cooled seem to get along better than those which have the cooling convectors and fans below the spaces to be cooled, and that it is far easier to blow warmed air down to the occupied spaces in winter than to blow cool air up to them in summer.

These observations and experiences led to an ambition in my office always to install an exhaust fan in cooperation with a supply fan on every cooling system, and to arrange interlocked dampers which will keep a definite relation between area of outside air intake, outboard exhaust and recirculation. They lead also to my advocacy of outside air intakes, free area through filters, convectors, ducts, etc., and outboard exhaust all capable of passing 100 per cent outside air. There are many days when the outside air condition is adequate to permit ventilation and comfort without any refrigeration, if only the plant is designed to take full advantage of the outside air. In a regrettable number of plants, because of failure to design for 100 per cent outside air, large power bills for refrigeration are incurred unnecessarily.

I designed a large general office cooling system in 1929. There were several private offices around the edges occupied by elderly executives. The central floor areas were crowded with employees, and electric lights were used all day long. I knew no better at that time than to supply the private offices from the ends of the concealed ducts which served the general spaces, and attempted to control the summer temperature in the private offices by throttling the air volume. Were I to design this plant today every private office would have a duct reheater under control of a room thermostat. By 1938, some of the occupants of the private offices were nine years more sensitive to drafts and asked me to do something about it. Duct reheaters could not be installed without tearing down the ceilings to get at the ducts, and the nuisance and cost would have been prohibitive. A few electric strip heaters placed behind the air inlet diffusers to the individual private offices and accessible through the diffusers solved this case admirably. On very hot days, when the entering air may be at 60 F or even cooler to care for the general offices, the strip reheaters go into action and add from 10 to 15 deg to the air temperature, and my old friends are happy.

"Finding the basement so comfortable and attractive, the owner proceeded to attract many more people to it."

Use of Forced Circulation Hot Water

Frequently I am called to diagnose and prescribe for ancient vapor heating systems in residences. The traps have been neglected, the alternating receivers or other appliances have worn out, and the air eliminators no longer eliminate, or if they do eliminate they also regurgitate. In many cases repairs for the obsolete equipment cannot be obtained. I approached a radical cure for the first one, some time ago, with trepidation, but now I tackle almost any one of them with confidence, and convert them to forced circulation hot water. In most cases no changes at all in the piping are necessary. Sometimes the old trap bodies can remain, with the interior mechanisms discarded and the valve seats removed or reamed larger.

It is comparatively easy to remove the steam trimmings and to install an expansion tank and automatic refill device with an electric water circulating pump. This converted system seems to give universal satisfaction. Perhaps it may not be so very much better than was the original vapor system when the latter was new, but it certainly is a far better heating plant than the limping, neglected vapor system.

I even had nerve enough to convert a 32 apartment four story steam system to forced circulation hot water, and have had a year's happy experience with its operation. This plant has a curious history. Originally it was a two pipe thermally circulating hot water system, with supply mains and an enormous expansion tank in the attic. The boiler was at one end of the building nearly a block from the other end and the plant evidently was sluggish in circulation. The apartments most distant from the boiler always were cold. From the mute evidence of its physical condition the original plant was very poorly designed and installed. After some years of unhappy experience the owner was advised by some unknown party to make a gravity steam job out of it. He installed hundreds of air valves and endured for many years the continual snapping and pounding, leaks and repairs inherent to a poorly installed and graded steam system. I found three old electric condensation pumps which had been installed and abandoned at various times in attempts by successive owners and contractors to correct the difficulties.

We installed a simple centrifugal water circulating pump, a large pneumatic expansion tank and made a very few changes in the basement mains. The return mains, long used for condensate which trickled along their bottoms, were channeled in some places as though the liquid they carried was strong acid. There were hardly any radiator valves which could be closed or opened, and all were minus the original handles. However, this plant, as a forced hot water system, is eminently satisfactory; it heats evenly and quietly and probably will last many years.

Secondary Air Filters

In a number of instances complaint is made of dust on walls and ceilings, which the owner thinks should have been prevented by the air cleaning devices. In several year around installations the dust has been traced—to a large extent, at least—to mud which accumulates on the dew-wetted summer cooling convectors, to leeward of the air filters. These convectors or evaporators dry off in winter and the dried debris continually flakes off and re-enters the air stream. This is an argument for secondary air filters, which indeed are being installed on many recent plants. No doubt but that much dust

too fine to be caught by the primary air filters "condenses" into larger and very objectionable particles as it passes along the ducts. With multiple cell type filter installations the owners are too prone to change all of the cells in any given unit when they change any of them. This of course greatly changes the resistance, increases the air volume, and encourages the carrying on of any dust which may have accumulated to leeward of the filters.

Miscellaneous Difficulties

Valves and Dampers—It is important that the automatic control valves on convectors which treat air from outside, and that the dampers which cut off the outside air, shall assume positive positions in case of failure of compressed air or electric energy. Very costly damage due to frozen convectors has been experienced due to closed supply valves or open dampers in emergencies of this type. Expansion and contraction of the ducts themselves have been proved to prevent the automatic closing of many a vitally important damper.

"There are many days when the outside air condition is adequate to permit ventilation and comfort without any refrigeration."

Antifreeze Mixtures—I would welcome more knowledge concerning antifreeze mixtures and a reduction in the cost of such compounds. In one case a large piping system which had been perfectly tight for months with water, leaked badly and disastrously when a common automobile antifreeze solution was administered.

Centrifugal refrigerating compressors have many admirable characteristics, especially along the line of flexibility and high efficiency under widely fluctuating loads. We need them in smaller sizes than the smallest now commercially available.

Finned tube convectors, wonderfully efficient as they are, have, it seems to me, in the stress of competition, been made with such small tubes, thin metal and fragile construction as to be too "flashy" and too likely to freeze. Rather than undergo the troubles which I have met from the clogged tiny tubes, the inability to drain out the water, and the high peaks and deep valleys of heat transmission rate, I would spend more money and devote more space for less efficient surfaces of much thicker metal holding much more water or steam or refrigerant.

Dampers—I am weary of looking through alleged tight dampers which show, against a light, crack leakages of relatively vast proportions. Any one who has worked on a sub-zero day behind the alleged protection of such dampers requires additional manually operated real doors at all outside air intakes; these doors being closed against weatherstripped jambs and being somewhat better than cobwebs for protection.

Trapping—I have just had an experience with a beautiful memorial theater finished in 1931. It is heated by high pressure steam from a central station. Fortunately, I was not the original designer. There are five separate steam supply circuits, each with a separate pressure reducing valve. The heat transmitters, which are radiators, thermally circulating enclosed convectors and finned copper blast convectors, and the ends of all supply mains, have thermostatic traps. The five separate return systems join a header near the condensation meter. There are no float traps on the ends of the return mains but all are dry, and the header is open to the atmosphere through a 1½ in. stand pipe. Every heat transmitter in the place has automatic temperature control. It is, of course, impossible to adjust the five reducing valves so that a uniform equal pressure on all of them is maintained, and the constant opening and closing of the diaphragm valves, changing the steam demand, would upset any attempt to maintain equal pressures in any event. I found the open vent discharging vapor and air, then sucking it in, in a constant breathing in and out, and I found some of the returns would be at subatmospheric pressure at the same time others were at pressures above atmospheric. Steam at 60 lb pressure is at about 306 deg temperature; when reduced to 2 lb pressure it will be superheated nearly 100 deg.

The owners gave a history of constant replacement of steam and return pipes. The public service company had paid for some of the replacement and had spent large sums of money analyzing the feedwater and the steam and the condensate all to no conclusion. Single pipe steam heating systems in the same city having ordinary air valves and float traps at the ends of the mains gave no trouble with corrosion.

My impression is that had each of the five separate circuits been provided with a float trap at the end of the respective return, with a non-return air vent on each, excess air under high temperature in the presence of condensate would have been so reduced as to cause no complaint about corrosion. Had there been a vacuum pump on the job, I think the re-entry of large quantities of air would have been prevented and that there would have been no trouble. Here there were significant factors in the superheat and in the wide pressure fluctuations in the returns due to complete automatic temperature regulation.

Fans—Intelligent manufacturers of centrifugal pumps have recognized and published the fact that the inlet and outlet pipe connections have a profound effect on efficiency. We always try to obtain at least five diameters of straight pipe approaching an inlet or an outlet. The same reasoning applies to the approach and discharge of a centrifugal fan. I blush, in the light of this obvious truth, when I think of the many short, abrupt fan connections of which I have been guilty. All double width, double inlet fan wheels should have rising pressure characteristics, as otherwise there will be competition between the two inlets, one or the other grabbing the load in succession, with chaotic pressures and eddies and unnecessary strain on the apparatus.

Centrifugal pumps always have some means for caring for the thrust endways of the shaft. This thrust is present in no small extent in single inlet centrifugal fans. Usually it is met by a shaft collar, subject to haphazard lubrication and rapid wear. Thrust collars therefore ought to be enclosed and lubricated with as much care as is given to the main bearings.

More attention should be given to fan selection with reference to electrical input. In a lamentable number of installations the motors are far oversize to care for load grabbing fans, with resultant inefficient load factor to be borne thereafter by the innocent owner.

From a Field Engineer's Notebook

Methods of supporting piping under various conditions that can't always be foreseen in time for the plans and must be determined after inspection of the situation in the field

By Allen M. Perry*

IN PLANNING construction or modernization of industrial and power plants, it is frequently impossible to indicate how all of the piping shall be supported due to various interferences which may arise. Frequently the type of supports must be determined by field engineers after inspecting the conditions.

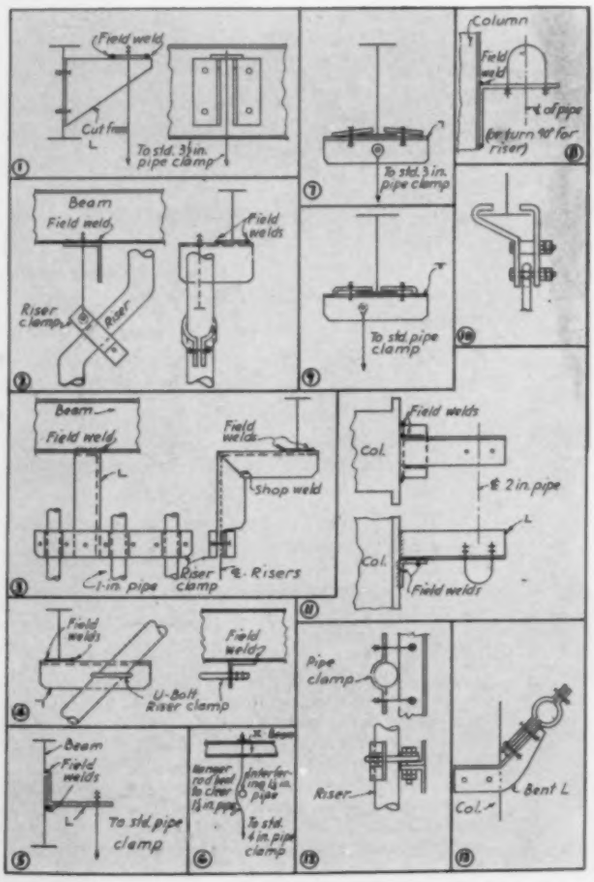
The accompanying sketches, which have been taken from field notebooks and drawings, indicate various ways in which different conditions were handled on a large modernization project. Their value is chiefly to expedite special designs that must be fabricated quickly in the field. Examination will indicate that standard beam and pipe clamps fit into many of them, and that simple combinations of structural shapes and the use of welding expedite construction and installation.

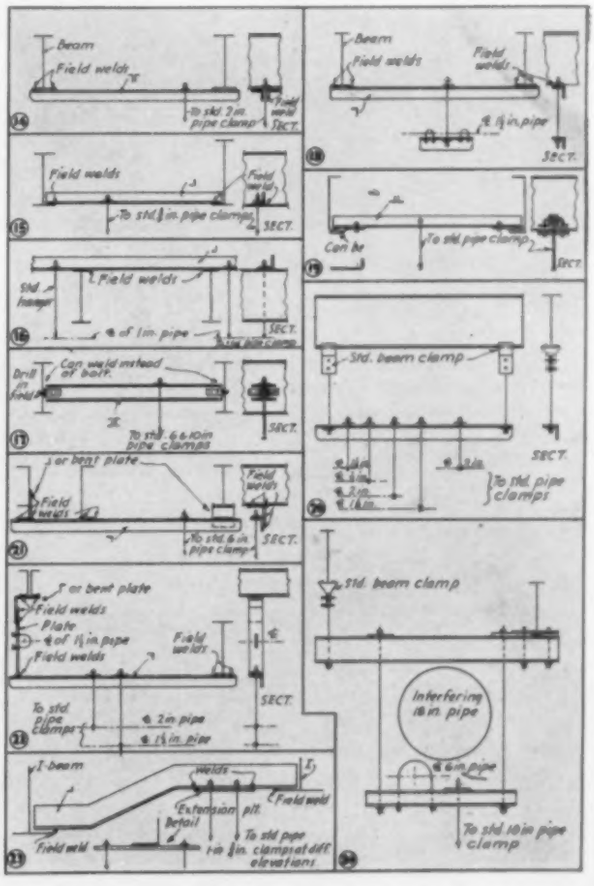
In December, 17 sketches appeared, and additional ones are to be published in a future issue.

*Formerly with M. H. Treadwell Co., General Contractor.

Key to Finding Pipe Support for Various Conditions

Condition	Sketch Numbers: Horizontal Runs	Risers
Between beams of same elevation	14-15-16-17-18-19	....
Between beams of different elevations	21-22-23-24	....
Near beam or platform	1-5-7-9-10-20	2-3-4
Near strut	.....	12
Near column	8-11-13	....
Interference above	6-24	2
Several pipes in same plane	16-18	3
Several pipes in different plane	20-22-23-24	....





Novel Scheme Employed for the Support of Heating Coil Units

By H. E. Ziel*

The only place available for installation of additional hot water heating coils, fans, motors, and filters at a one story automobile plant was the roof. As the roof trusses were already heavily loaded, the added loads were imposed directly on the building columns. . . . In one case, the load was placed eccentric from the center of the girder carrying the heating equipment, and the small upward force created by the eccentricity was delivered to the old truss. This actually decreased the original load on the truss, instead of adding to it

THE best location for installing heating, ventilating or air conditioning equipment in an existing building is many times determined by a study of the structural details. A most unusual problem was solved in a simple manner at the Plymouth Motor Car Co. plant in Detroit several years ago, when it was found necessary to supply 150,000 cfm to certain areas.

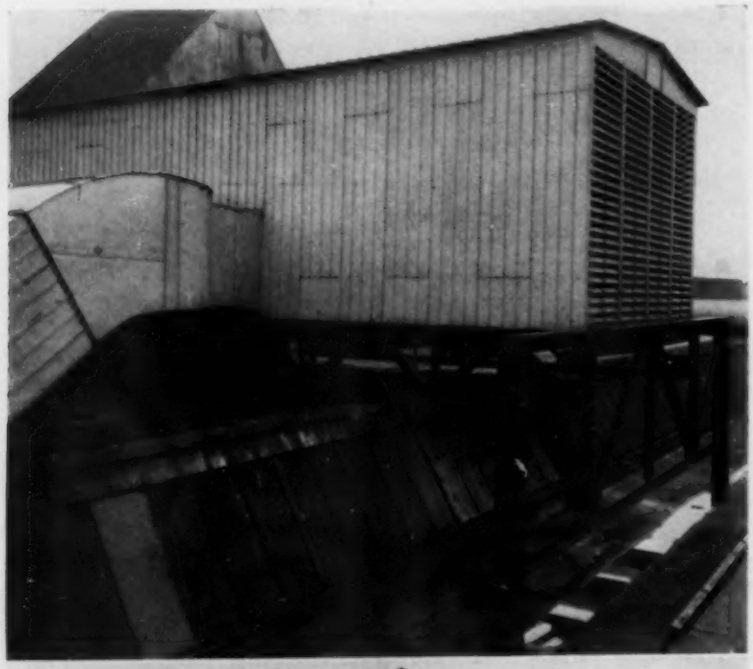
The particular building is one story in height with practically every square foot of floor space used for productive work. The only place available for installing the required fans, motors, heaters and air filters was on the roof. The roof trusses were already heavily loaded with piping, conveyors, etc. The heating medium is hot water, heated by the exhaust system from drop hammers some 2000 ft away. Part of the hot water is used through direct radiation to heat various areas in the building. To balance the existing hot water distribution system, it was necessary to select heating coils that would be (1) satisfactory for hot water and (2) capable of allowing the required amount of hot water to pass through the coils at a minimum pressure drop.

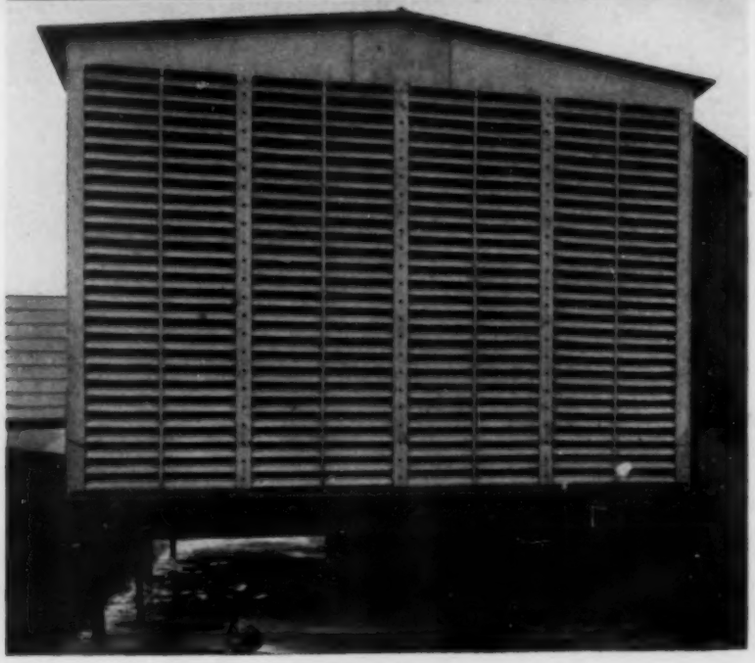
Three 50,000 Cfm Units

It was decided to divide the required capacity into three 50,000 cfm units. Inasmuch as the roof trusses were already overloaded, it was planned to impose the loads directly onto the building columns, placing the equipment with their enclosing houses above the roof.

The columns of the original steel frame together with their foundations chanced to be strong enough to carry the additional fan house loads, but (as mentioned) the trusses and roof beams could not handle the extra loads. As a consequence, it was decided to extend the columns supporting two of the fan houses above the top of the "A" frame and provide trusses and beams over the "A" frame to carry the equipment and housings. Because four columns were available this installation proved simple. (See Fig. 1).

However, in the other case where the equipment was situated in a narrow valley and only





*Albert Kahn, Inc., Architects and Engineers.

Fig. 1 (above)—In the case of two of the heating coil and fan houses, four columns were extended above the top of the "A" frame, and trusses and beams were provided to carry the heating and ventilating equipment. . . . Fig. 2 (below)—In the other case, only two columns could be used, so the supporting beams were set athwart a girder and nearly balanced on it. The load was placed eccentric for stability, and the small upward force thus created was delivered to the roof truss, which decreased the original load

two columns could be utilized, a very unusual type of support was devised, as shown in Fig. 2. The two columns were first extended up through the old roof. To the column extensions a 30 in. girder was framed, Fig. 3, and beyond the ends of this girder, angle brackets were provided to extend the platform beyond the ends of the girder as required.

This done, seven 10 in. beams were set athwart the girder and nearly balanced on it, the idea being to balance the fan and equipment on the girder. However, to insure stability, the center of gravity of the fan, etc., load was deliberately placed 12 in. eccentric from the center of the girder, creating a small upward tension at one end of the 10 in. beams. This end of all the 10 in. beams was then attached to a 16 in. beam. The ends of the 16 in. beam, in turn, were secured to a vertical tie made of two angles, Fig. 4, the lower end of which was fastened to the old roof trusses below.

The load of fan and heater, located eccentrically as already mentioned, was therefore supported entirely by the 30 in. girder and the small *upward* force created by the eccentricity was delivered to the old truss.

This upward force, instead of adding to the original load, actually decreased it.

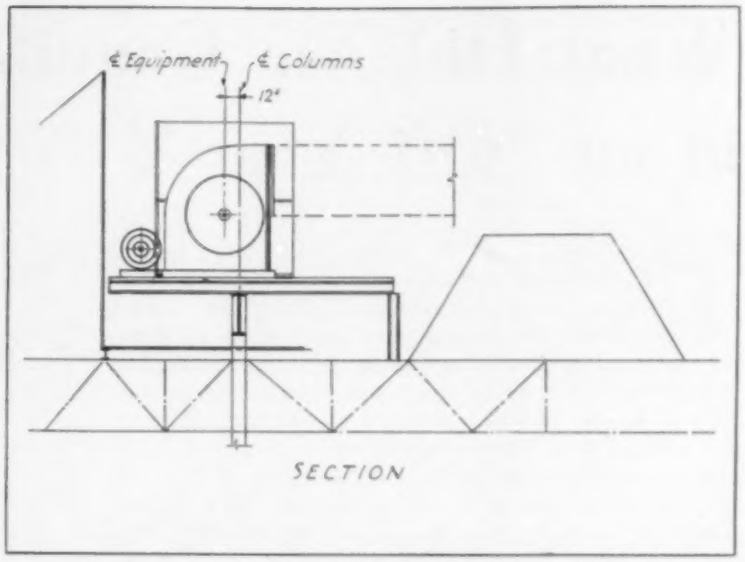
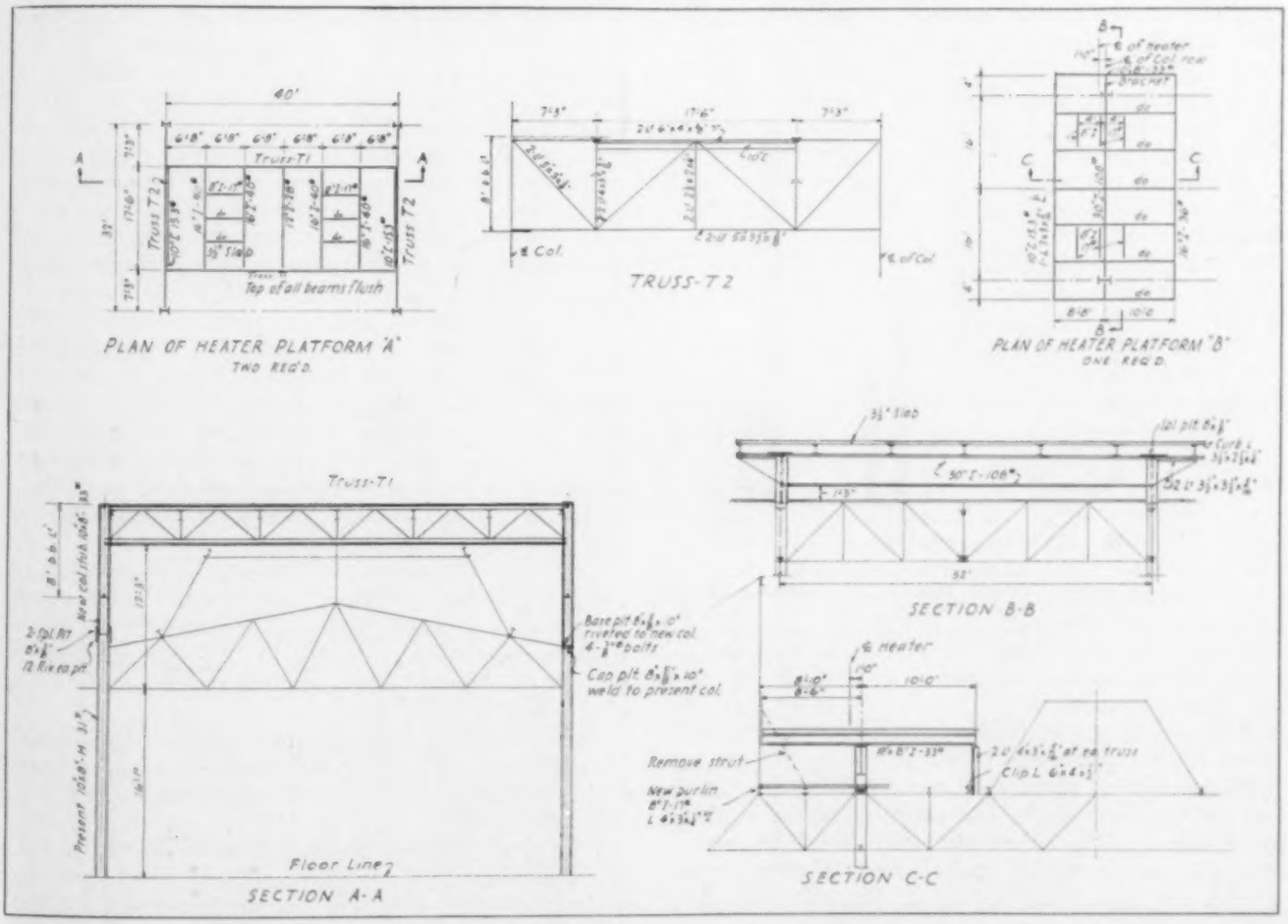


Fig. 3—Detail of view shown in Fig. 2. A 30 in. girder was framed to the column extensions

Fig. 4—Details of supporting structure for case shown in Fig. 2



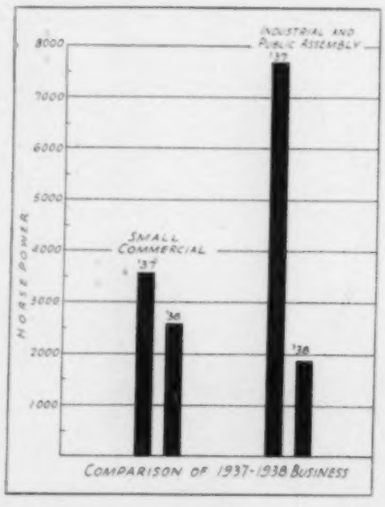
What Did Air Conditioning Do in an "Off Year"?

Knight Porter* and William Rock* analyze the records of air conditioning installations in Chicago during 1937 and 1938, show that while the total decrease ('38 compared with '37) was 18 per cent in number of installations, and 60 per cent in connected horsepower, the corresponding percentages for the small, highly competitive types of buildings were but 9 and 26 per cent, respectively. . . . Their analysis indicates that where the motive for buying air conditioning is profit and local competition, the volume holds up very well despite poor general business conditions. Where the buying motive is largely comfort and efficiency (human or industrial)—that is, non-competitive—general business conditions have much more effect on the volume of air conditioning installations. However, this is felt to be a postponement of air conditioning investment, and with returning business confidence the jobs that didn't go in in 1938 will be installed

THE history of the volume of the air conditioning business in most communities has been one of continual increase from year to year. Such has been the case in Chicago during the past six years—with the exception of 1935, which was not quite able to keep pace with the 1934 volume, stimulated as the latter was by the Century of Progress exposition.

There is little question that 1938 was an "off year" for air conditioning just as it was for general business. The Chicago horsepower in 1938 returned to the 1935 level. Whereas formerly we have been analyzing markets from an increase viewpoint, we may now study the source and extent of decrease. We may also want to know what the air conditioning markets were that held up well in a poor business year.

The overall decrease in Chicago air conditioning volume (as measured in horsepower) for the first 11 months of 1938 was approximately 60 per cent as compared with 1937. Recognizing that the 1938 business was typified by a good volume in the small, highly competitive retail stores and almost a complete absence of large office installations, we decided to attempt to find a common ground on which to divide the market analysis. The first group (see table) we have called "Small Commercial." The buying motive in this group has been profit and local competition; many of these buyers of air conditioning could not afford to be without it. It will be noticed that this group as a whole shows for 1938 only a 9 per cent decrease in the number of installations and a 26 per cent decrease in horsepower as compared with the similar 1937 period. It will be noticed that the following markets showed an improvement in 1938: beauty shops, bowling alleys, funeral parlors, candy stores, clothing stores, drug stores, food stores, and photographic studios.



Although non-competitive air conditioning installations decreased 35 per cent in number and 76 per cent in horsepower last year compared with 1937, the competitive types of jobs decreased but 9 and 26 per cent

We have called the second group "Industrial, Office and Public Assembly." The majority of the markets under this heading are non-competitive. The buying motive is largely comfort and efficiency (human or industrial). There are two markets, hotels and theaters, included here which must be separately explained. One large hotel installation of more than 1000 hp is included in the 1937 hotel group. A major portion of this installation was for guest rooms, and since there were practically no other hotels in the city with air conditioned guest rooms it can hardly be said that this installation was influenced by strictly competitive interests. Theaters are also included in this second group on the basis of the high degree of saturation in this market. Sixty per cent of the listed theaters (322) are now conditioned; those remaining without air conditioning are highly marginal houses which would require extensive remodelling as well as air conditioning were they to be modernized. For this reason, it is felt that the theater market should be excluded from the highly competitive "Small Commercial" group. It will be noticed in this second group that the number of installations decreased 35 per cent, while the horsepower connected decreased 76 per cent in 1938 as compared with 1937. Also, the average size of installation decreased from 62.7 hp in 1937 to 23 hp in 1938.

The most notable loss in 1938 is to be found in large installations. In 1938 there were but five installations of 100 tons or larger as compared with 14 in 1937. There were no 1938 contracts for 200 tons and larger installations, whereas there were five installations in 1937 of 200 tons or more.

General offices as a classification showed the greatest loss, 2621 hp. The 1937 business produced seven general office jobs of 100 tons and larger as compared with only two in 1938.

One of the highest percentage losses was in industrial applications. This loss was 53 per cent in number of installations and 82 per cent in horsepower. This

*Air Conditioning Div., Commonwealth Edison Co.

was to be expected because of the low level of general business and industrial activity.

In summary, let it be mentioned that in the light of 1938 general business conditions, small commercial air conditioning made a very good record; the retarded activity in commercial remodelling and new building is the only factor which prevented this group from bettering the 1937 record. Large installations (the second classification) suffered directly with the low ebb of business. Reduced volume apparently decreased interest in production efficiency and therefore in industrial air conditioning, and corporation profits were such as to cause postponement of office modernizing and air conditioning.

Since air conditioning has suffered only postponement, it may be said that no ground has been lost. With the return of business confidence we may look to future volume which will be greater than past levels to the extent of 1938 postponements in investing in air conditioning.

1937-38 Air Conditioning Volume Compared

(Selected types of installations for 11 month periods for Chicago)

	1937		1938	
Small Commercial	No. of Jobs	Horsepower	No. of Jobs	Horsepower
Barber Shops	1	1	..	..
Beauty Shops	5	19	8	33
Bowling Alleys	1	33	3	107
Funeral Parlors	15	224	20	173
Restaurants	98	1821	87	1325
Stores				
Candy	7	28	14	73
Clothing, Dept.	18	442	24	449
Drug	21	187	21	197
Food	5	31	8	32
Fur	6	30	3	16
Shoes	27	344	9	66
Miscellaneous	21	406	6	115
Studios	3	10	4	52
Total	228	3576	207	2638
Average Installation..		15.7		12.7
Per Cent Decrease: No. of Installations, 9.		Horsepower, 26.		
Industrial, Office, and Public Assembly				
Banks	..	..	2	38
Churches	..	..	3	128
Civic Buildings	..	..	1	30
Clubs	4	37	..	..
Hospitals	5	5	2	7
Hotels	5	1117	3	45
Industrial				
Bakery	1	4	1	15
Candy	4	32	2	23
Printing	6	462	2	47
Miscellaneous	6	626	3	111
General Offices	59	3585	55	964
Theaters	33	1847	6	437
Total	123	7715	80	1845
Average Installation..		62.7		23
Per Cent Decrease: No. of Installations, 35.		Horsepower, 76.		
GRAND TOTAL	351	11,291	287	4483
Per Cent Decrease: No. of Installations, 18.		Horsepower, 60.		

Note—Room coolers have not been included in this 11 month tabulation. Small residential and room cooler sales have progressed each year, and the development of the art is such as to indicate an accelerated rate of sales with improved general business conditions.

Heating, Piping and Air Conditioning's Annual Air Conditioning Review

ON THE next page are tabulated comparative data on the number of installations and connected horsepower for air conditioning commercial and institutional buildings, industrial plants, and residences for the full year of 1937 and the first 11 months of 1938, with totals as of December 1, 1938. These figures were obtained from the utility companies serving the various areas covered, and the courtesy of the utilities in making the data available is gratefully acknowledged.

If the figures for those cities for which the necessary data are available are totalled, it will be found from the table that the number of air conditioning installations during the first 11 monhs of last year was greater than in 1937, but that the horsepower for the 1938 installations was considerably below that of the 1937 jobs. For cities and areas representing between 29 and 30 million population and listed in the table on the next page, the figure for number of 1937 installations is 4,963; for 11 months of 1938 the corresponding figure is 5,322. The 4,963 1937 jobs represented 92,496 connected horsepower, however, while the 5,322 jobs during the first 11 months of 1938 accounted for but 60,956 hp.

An analysis of 1938 air conditioning volume as compared with that of 1937 is given in the article by Porter and Rock immediately preceding. It is based on the figures for Chicago, where complete data are available.

Data such as are given in the table on the following page are still difficult to compile with absolute accuracy, and cannot be obtained for as many cities as would be desirable. The information can be had only from electric utilities, and therefore (in many cases) the figures given do not cover installations which are not served by the utility—such as many industrial jobs, and jobs using other forms of refrigeration or latent heat removal than electric motor driven. Thus, horsepower figures do not fully reflect the installed air conditioning capacities. In the tabulation, the figures do not include installations not served by the utility unless marked (†). The horsepower figures shown include the connected hp for fans and pumps, as well as compressors, unless marked (*). "N.R." means "no record received." In one or two cases, the 1938 figures include only 10 instead of 11 months.

While the scope and form of the reported data vary for each reporting utility, the comparison between 1937 and 1938 for any one city is felt to be accurate, for both sets of figures were of course obtained from the same source.

The American Ice Co. (serving metropolitan New York and Boston, and the cities of Philadelphia, Washington, and Baltimore) reports the following numbers of new ice using air conditioning installations in its territories during 1938—legitimate theaters, 6; motion picture theaters, 3; restaurants, 15, and miscellaneous, 6. The company's total air conditioning ice consumption for the 1938 season (old jobs as well as new, of course) was 105,075 tons—low because of weather conditions. Few of the installations in the company's territories required any refrigeration at all prior to July 15.

It is planned to publish further data on 1938 air conditioning installations—including figures on railway air conditioning—as they become available later on.

Number of Air Conditioning Installations and Connected Horsepower for Air Conditioning for Commercial and Institutional Buildings, Industrial Plants, and Residences for 1937, 1938, and Totals as of Dec. 1, 1938

Area	1937								1938 (11 Months)								Totals as of Dec. 1, 1938								Approx. Population of Area
	All Buildings		Com. & Inst.		Industrial		Residential		All Buildings		Com. & Inst.		Industrial		Residential		All Buildings		Com. & Inst.		Industrial		Residential		
	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	
ALABAMA																									
Birmingham†	61	1,655	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	39	379	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	190	4,475	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	300,000
CALIFORNIA																									
So. Calif. Ed. Co.	280	2,960	220	1,024	26	1,871	34	65	242	1,214	194	655	21	447	27	112	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	.....
DIST. OF COLUMBIA																									
Washington	218	6,451.5	182	6,340.75	1	68	35	42.75	189	4,125.75	159	3,432	6	307	24	386.75	911	50,019.5	738	48,885.5	15	531	158	603	627,000
IDAHO																									
Idaho Power Co.	47	134	46	133	.....	.....	1	1	50	100	50	100	.....	.....	.....	.....	208	632	183	621	.....	.....	25	11	200,000
ILLINOIS																									
Chicago†	575	12,556	500	11,197	18	1,269	57	90	660	4,819	541	4,511	8	196	111	112	2,583	75,026	2,150	64,829	149	9,828	284	369	3,500,000
Rockford	46	273.18	28	257.78	.....	.....	18	15.4	33	108.75	23	101.5	.....	.....	10	7.25	104	814.68	62	769.78	.....	.....	42	44.9	85,000
INDIANA																									
Indianapolis and Marion Co.†	113	1,259.2	87	1,213.7	3	22	23	23.5	67	711.51	53	632.17	3	66.5	11	12.84	347	6,706.21	256	5,978.37	12	601	79	126.84	400,000
IOWA																									
Cedar Rapids	21	304.5	19	300	.....	.....	2	4.5	24	235.5	22	233.75	.....	.....	2	1.75	93	1,255	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	60,000
Des Moines	97	1,773.61	76	1,739.38	.....	.....	21	34.23	65	435.76	51	417.94	.....	.....	14	17.82	316	5,617.03	245	5,474.05	5	42.6	66	100.38	150,000
KANSAS																									
Kansas G. & E. Co.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	66	504.5	43	464.1	.....	.....	23	40.4	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	.....
KENTUCKY																									
Louisville	89	1,495.54	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	48	354.97	46	346.97	2	8	.....	.....	262	4,424.92	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	350,000
MARYLAND																									
Baltimore and Vicinity	199	3,435	132	3,232	4	131.5	63	71.5	235	3,097	152	2,941	3	55	80	101	778	16,187.5	566	15,115	20	800	192	272.5	1,160,000
MASSACHUSETTS																									
Boston Edison Co.	247	1,716	216	1,485	6	197	25	34	108	1,414	97	1,344	8	60	3	10	591	14,609	482	13,309	35	1,152	74	148	1,400,030
MICHIGAN																									
Detroit and Vicinity*†	336	4,120	201	3,844.25	8	129.5	127	146.25	182	3,255.57	142	2,547.57	8	661.5	32	46.5	1,008	27,835.81	656	24,287.81	41	3,135	311	413	2,540,000
Consumers Power Co.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	81	724.15	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	.....
MISSOURI																									
St. Louis (Greater)	588	11,046.29	315	9,861.76	18	650.28	255	534.25	492	3,857.2	256	3,481.95	5	71.84	231	303.41	2,051	36,988.85	1,102	32,622.5	67	2,781.56	882	1,584.79	1,250,000
NEW HAMPSHIRE†	8	47.25	5	36.25	1	7.5	2	3.5	23	102.75	20	97.75	1	3.75	2	1.25	43	235.5	33	193.75	6	37	4	4.75	270,000
NEW YORK																									
Ithaca	12	87.5	12	87.5	.....	.....	.....	.....	4	81	4	81	.....	.....	.....	.....	16	168.5	16	168.5	.....	.....	.....	.....	22,000
New York City	415	16,203.5	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	376	16,461.8	363	15,791.97	6	661.5	7	8.33	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	6,700,000
Rochester	50	2,409	33	442.5	8	1,952.25	9	14.25	33	322	27	295.75	4	21.5	2	4.75	162	29,589	110	4,015	20	25,516.25	32	57.75	469,000
Syracuse*	63	1,803.28	53	1,562.03	6	230.5	4	10.75	93	433.5	78	208	1	215	14	10.5	156	2,236.78	131	1,770.03	7	445.5	18	21.25	235,000
Watertown	4	24	1	15	3	9	.....	.....	3	20	1	5	2	15	.....	.....	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	40,000
OHIO																									
Akron	21	455.04	20	452.04	.....	.....	1	3	8	121	7	119.5	.....	.....	1	1.5	75	1,467.44	67	1,458.69	.....	.....	8	10.75	350,000
Dayton	73	1,131.15	35	904.34	16	192.99	22	33.82	51	713.28	29	641.08	1	55	21	17.2	240	4,258.5	124	3,156	44	999	72	103.5	375,000
Toledo†	40	432	22	198	8	199	10	35	40	225	29	211	.....	.....	11	14	179	3,833	128	3,150	16	611	35	72	300,000
OKLAHOMA																									
Okla. G. & E. System	194	2,100	98	1,864	5	24	91	212	177	1,950	102	1,765	.....	.....	75	185	662	8,027	357	7,345	9	45	296	637	590,800
PENNSYLVANIA																									
Pa. P. & L. Co. System	36	1,051	27	760	4	276	5	15	40	2,182	37	1,912	3	270	N.R.	N.R.	192	7,241	171	5,666	21	1,575	N.R.	N.R.	1,680,000
Philadelphia Area	257	8,657	183	7,310.7	16	1,279	58	67.3	290	7,872.1	200	6,583.2	9	1,164.5	81	124.4	1,133	42,694.1	771	34,396.3	119	7,904.6	243	393.2	2,754,000
Pittsburgh†	212	4,095.25	165	3,988	4	49.5	43	57.75	211	1,406.75	172	1,329.25	5	51.25	34	26.25	726	13,476	563	12,570.5	22	732.25	141	173.25	1,400,000
RHODE ISLAND																									
Narragansett Elec. Co., Providence	22	344	21	336.5	.....	.....	1	7.5	16	176.25	13	168	.....	.....	3	8.25	79	1,708	70	1,684	.....	.....	9	24	470,000
SOUTH CAROLINA																									
Charleston	11	170.75	9	163	.....	.....	2	7.75	3	167.5	3	167.5	.....	.....	.....	.....	20	688.25	18	680.5	.....	.....	2	7.75	65,000
TENNESSEE																									
Chattanooga	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	13	141	N.R.	N.R.	4	3	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	N.R.	150,000
TEXAS																									
Beaumont & Port Arthur	17	362.7	16	360	.....	.....	1	2.7	26	603	14	577	2	10	10	16	69	1,865.3	52	1,756.3	3	18	14	91	115,000
El Paso	464	545	14	430	.....	.....	450	115	1,359	944	9	594	.....	.....	1,350	350	1,887	1,936	87	1,471	.....	.....	1,800	465	110,000
San Antonio	53	2,309.25	41	2,281	.....	.....	12	28.25	51	1,485	30	1,417	.....	.....	21	68	221	10,176.5	153	10,018.5	.....	.....	68	158	232,000
VIRGINIA																									
Richmond District	37	388	32	381	.....	.....	5	7	52	1,099	45	832	1	260	6	7	203	5,891	152	4,914	13	910	38	67	390,000
Va. P. S. Co. System	51	572.3	51	572.3	.....	.....	.....	.....	27	239.6	27	239.6	.....	.....	.....	.....	107	911.77	107	911.77	.....	.....	.....	.....	448,500
WISCONSIN																									
Racine County	6	128.75	3	7.25	1	120	2	1.5	5	242.5	.....	.....	2	240	3	2.5	11	371.25	3	7.25	3	360	5	4	72,000

[See text on preceding page]

Properties of Air-Vapor Mixtures Interpolated to Tenths of a Degree

By William Goodman*

COLUMNS 2 and 5 of the accompanying tables (next page) have been interpolated from the Psychrometric Tables that appeared in the January, 1938, issue.[1] Although it is doubtful whether the various quantities in the table are known accurately enough to be stated definitely to the nearest tenth of a degree, nevertheless such tables are extremely useful in air conditioning work. Cases frequently arise where values to tenths of a degree must be used if consistent results are to be obtained from a set of computations. For instance, in making curves, the points may scatter unless values interpolated to tenths of a degree have been used throughout the computations. Furthermore, in much air conditioning work the air may be cooled through only a small wet bulb temperature range. In these circumstances the amount of heat removed will be small. Therefore, unless values of enthalpy to at least one-fifth of a degree are used, a relatively large percentage of error may result.

The values in Columns 3 and 4 are extremely handy for quickly finding the *actual* enthalpy (total heat) of an air-vapor mixture if its dry bulb and dew point temperatures are known. A complete discussion was given in the January, 1938, article of the method of using the actual enthalpy of the air for accurate computations as against the approximate methods commonly used. Although the error is generally not large, there are times when accurate computations using the actual enthalpy should be used. In such cases the equation below, which was developed in the earlier article, offers the quickest means of finding the actual enthalpy if the dry bulb and dew point temperatures are known.

$$h = s'' t + 1061w$$

where h = actual enthalpy of the mixture, Btu per lb of dry air;
t = dry bulb temperature of the air.

Values of s'', the humid specific heat of air corresponding to any given dew point temperature, are tabulated in Column 3. Values of $1061w$ corresponding to the given dew point temperature are tabulated in Column 4.

[The tables will be continued next month]

*The Trane Co. Member of Board of Consulting and Contributing Editors.

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[1] "New Tables of the Psychrometric Properties of Air-Vapor Mixtures," HEATING, PIPING AND AIR CONDITIONING, January, 1938, pp. 1-7.

Portable Air Conditioner Installed on Ship

Results of the first installation of a portable air conditioner on a ship, the Dutch Mailship "Marnix van St. Aldegonde" which carried the Sultan van Langkat of the Dutch East Indies to Holland to assist at the 40-Year Jubilee of Queen Wilhelmina, are being studied by marine engineers for future application of this science for sea transportation. The unit which was installed in the Sultan's de luxe cabin is of the water cooled type. Fresh water was used for the condenser, as a special condenser for sea water was not available when this installation was made.

The air conditioning men had an opportunity to inspect the equipment on a trial run from the Dutch East Indies to Holland and found that remarkably few alterations had to be made in spite of heavy seas.

Compressor motor, fan motor, fan and housing and electrical controls were changed for 220 volt direct current. The discharge pipe of the compressor with the flexible joint was relocated so that it would not strike any part of the unit, while in a still or rolling position. An arrangement was made so that only one line and one hole had to go through the side of the ship for the common drain line of the evaporator and the outlet of the condenser.

A third mounting was installed under the primary air fan because of the rolling of the ship but the spring mountings under the compressor weathered the trip in good shape. As windows and shutters on the ship are lowered when opening, a small wooden piece was placed in the top part of the window frame, and the outside air duct connected to this board and the back of the unit. This allowed full use of the windows and shutters without extensive alterations.

When the unit is removed, only four wooden plugs will have to be taken out and the window sill refinished where they were installed.—Gerald Burke, Carrier Corp.

Trend to Welded Valve Bonnets Seen at New York Power Show

Hard faced discs and seats, welding ends and chrome plated stems are now accepted construction for high temperature-pressure valves; but at the recent power show in New York City there was also a noticeable trend toward welded bonnets. Evidently materials have been perfected to a degree which obviates concern for valve maintenance, the advantages of welded construction being considered to outweigh any handicap due to non-accessibility resulting from welded bonnets.

The welding end valves at the power show were threaded (either internally or externally), or else provided with special collars. These constructions have been adopted in order that valves may be tested without undue stresses such as the compressive force of a yoke to seal the valve ends.

Heating-Piping

Absolute Humidity and Enthalpy of Mixtures of Air and Saturated Water Vapor

Interpolated to Tenths of a Degree.
Barometric Pressure 14.696 Lb per Sq In.
All Tabulated Values are Quantities per Pound of Dry Air in the Mixture.

Temp. Deg Fahr t	Absolute Humidity Pounds w	Humid Specific Heat s'	$1061w$	Enthalpy of Air-Vapor Mixture Btu h	Temp. Deg Fahr t
1	2	3	4	5	6
−20.0	0.0003481	0.2402	0.3693	−4.434	−20.0
.9	.0003500		.3713	−4.408	.9
.8	.0003518		.3733	−4.382	.8
.7	.0003537		.3753	−4.356	.7
.6	.0003556		.3773	−4.330	.6
−19.5	0.0003575	0.2402	0.3793	−4.304	−19.5
.4	.0003593		.3812	−4.278	.4
.3	.0003612		.3832	−4.252	.3
.2	.0003631		.3852	−4.226	.2
.1	.0003649		.3872	−4.200	.1
−19.0	0.0003668	0.2402	0.3892	−4.174	−19.0
.9	.0003687		.3912	−4.148	.9
.8	.0003706		.3933	−4.122	.8
.7	.0003726		.3953	−4.096	.7
.6	.0003745		.3974	−4.070	.6
−18.5	0.0003764	0.2402	0.3994	−4.044	−18.5
.4	.0003783		.4014	−4.018	.4
.3	.0003802		.4035	−3.992	.3
.2	.0003822		.4055	−3.966	.2
.1	.0003841		.4076	−3.940	.1
−18.0	0.0003860	0.2402	0.4096	−3.914	−18.0
.9	.0003880		.4118	−3.888	.9
.8	.0003901		.4139	−3.862	.8
.7	.0003921		.4161	−3.835	.7
.6	.0003941		.4182	−3.809	.6
−17.5	0.0003962	0.2402	0.4204	−3.783	−17.5
.4	.0003982		.4225	−3.757	.4
.3	.0004002		.4247	−3.731	.3
.2	.0004022		.4268	−3.704	.2
.1	.0004043		.4290	−3.678	.1
−17.0	0.0004063	0.2402	0.4311	−3.652	−17.0
.9	.0004085		.4334	−3.626	.9
.8	.0004106		.4357	−3.599	.8
.7	.0004128		.4379	−3.573	.7
.6	.0004149		.4402	−3.547	.6
−16.5	0.0004171	0.2402	0.4425	−3.521	−16.5
.4	.0004192		.4448	−3.494	.4
.3	.0004214		.4471	−3.468	.3
.2	.0004235		.4493	−3.442	.2
.1	.0004257		.4516	−3.415	.1
−16.0	0.0004278	0.2402	0.4539	−3.389	−16.0
.9	.0004300		.4562	−3.363	.9
.8	.0004322		.4586	−3.336	.8
.7	.0004344		.4609	−3.310	.7
.6	.0004366		.4632	−3.284	.6
−15.5	0.0004388	0.2402	0.4656	−3.258	−15.5
.4	.0004410		.4679	−3.231	.4
.3	.0004432		.4702	−3.205	.3
.2	.0004454		.4725	−3.179	.2
.1	.0004476		.4749	−3.152	.1
−15.0	0.0004498	0.2402	0.4772	−3.126	−15.0
.9	.0004521		.4797	−3.100	.9
.8	.0004545		.4822	−3.073	.8
.7	.0004568		.4846	−3.047	.7
.6	.0004591		.4871	−3.020	.6
−14.5	0.0004615	0.2402	0.4896	−2.994	−14.5
.4	.0004638		.4921	−2.967	.4
.3	.0004661		.4946	−2.941	.3
.2	.0004684		.4970	−2.914	.2
.1	.0004708		.4995	−2.888	.1
−14.0	0.0004731	0.2402	0.5020	−2.861	−14.0
.9	.0004756		.5046	−2.834	.9
.8	.0004780		.5072	−2.808	.8
.7	.0004805		.5098	−2.781	.7
.6	.0004829		.5124	−2.755	.6
−13.5	0.0004854	0.2402	0.5151	−2.728	−13.5
.4	.0004879		.5177	−2.701	.4
.3	.0004903		.5203	−2.675	.3
.2	.0004928		.5229	−2.648	.2
.1	.0004952		.5255	−2.622	.1
−13.0	0.0004977	0.2402	0.5281	−2.595	−13.0
.9	.0005002		.5307	−2.568	.9
.8	.0005028		.5335	−2.542	.8
.7	.0005053		.5362	−2.515	.7
.6	.0005079		.5389	−2.488	.6
−12.5	0.0005104	0.2402	0.5416	−2.462	−12.5
.4	.0005129		.5442	−2.435	.4
.3	.0005155		.5469	−2.408	.3
.2	.0005180		.5496	−2.381	.2
.1	.0005206		.5523	−2.355	.1
−12.0	0.0005231	0.2402	0.5550	−2.328	−12.0
.9	.0005257		.5578	−2.301	.9
.8	.0005284		.5606	−2.274	.8
.7	.0005310		.5634	−2.248	.7
.6	.0005336		.5662	−2.221	.6

Temp. Deg Fahr t	Absolute Humidity Pounds w	Humid Specific Heat s'	$1061w$	Enthalpy of Air-Vapor Mixture Btu h	Temp. Deg Fahr t
1	2	3	4	5	6
−11.5	0.0005363	0.2402	0.5690	−2.194	−11.5
.4	.0005389		.5717	−2.167	.4
.3	.0005415		.5745	−2.140	.3
.2	.0005441		.5773	−2.114	.2
.1	.0005468		.5801	−2.087	.1
−11.0	0.0005494	0.2403	0.5829	−2.060	−11.0
.9	.0005522		.5859	−2.033	.9
.8	.0005551		.5889	−2.006	.8
.7	.0005579		.5920	−1.979	.7
.6	.0005608		.5950	−1.952	.6
−10.5	0.0005636	0.2403	0.5980	−1.925	−10.5
.4	.0005664		.6010	−1.898	.4
.3	.0005693		.6040	−1.871	.3
.2	.0005721		.6071	−1.844	.2
.1	.0005750		.6101	−1.817	.1
−10.0	0.0005778	0.2403	0.6131	−1.790	−10.0
.9	.0005807		.6162	−1.763	.9
.8	.0005836		.6192	−1.736	.8
.7	.0005865		.6223	−1.709	.7
.6	.0005894		.6253	−1.682	.6
− 9.5	0.0005923	0.2403	0.6284	−1.655	− 9.5
.4	.0005951		.6315	−1.627	.4
.3	.0005980		.6345	−1.600	.3
.2	.0006009		.6376	−1.573	.2
.1	.0006038		.6406	−1.546	.1
− 9.0	0.0006067	0.2403	0.6437	−1.519	− 9.0
.9	.0006098		.6470	−1.492	.9
.8	.0006129		.6503	−1.464	.8
.7	.0006160		.6535	−1.437	.7
.6	.0006191		.6568	−1.410	.6
− 8.5	0.0006222	0.2403	0.6601	−1.383	− 8.5
.4	.0006252		.6634	−1.355	.4
.3	.0006283		.6667	−1.328	.3
.2	.0006314		.6699	−1.301	.2
.1	.0006345		.6732	−1.273	.1
− 8.0	0.0006376	0.2403	0.6765	−1.247	− 8.0
.9	.0006407		.6798	−1.219	.9
.8	.0006439		.6832	−1.191	.8
.7	.0006470		.6865	−1.164	.7
.6	.0006502		.6898	−1.137	.6
− 7.5	0.0006533	0.2403	0.6932	−1.109	− 7.5
.4	.0006564		.6965	−1.082	.4
.3	.0006596		.6998	−1.054	.3
.2	.0006627		.7031	−1.027	.2
.1	.0006659		.7065	−0.9997	.1
− 7.0	0.0006690	0.2403	0.7098	−0.9723	− 7.0
.9	.0006723		.7133	−0.9448	.9
.8	.0006756		.7168	−0.9172	.8
.7	.0006789		.7203	−0.8897	.7
.6	.0006822		.7238	−0.8622	.6
− 6.5	0.0006856	0.2403	0.7274	−0.8347	− 6.5
.4	.0006889		.7309	−0.8071	.4
.3	.0006922		.7344	−0.7796	.3
.2	.0006955		.7379	−0.7521	.2
.1	.0006988		.7414	−0.7245	.1
− 6.0	0.0007021	0.2403	0.7449	−0.6970	− 6.0
.9	.0007056		.7486	−0.6692	.9
.8	.0007091		.7524	−0.6415	.8
.7	.0007127		.7561	−0.6137	.7
.6	.0007162		.7599	−0.5860	.6
− 5.5	0.0007197	0.2403	0.7636	−0.5582	− 5.5
.4	.0007232		.7673	−0.5304	.4
.3	.0007267		.7711	−0.5027	.3
.2	.0007303		.7748	−0.4749	.2
.1	.0007338		.7786	−0.4472	.1
− 5.0	0.0007373	0.2403	0.7823	−0.4194	− 5.0
.9	.0007409		.7861	−0.3915	.9
.8	.0007445		.7900	−0.3637	.8
.7	.0007481		.7938	−0.3358	.7
.6	.0007517		.7976	−0.3080	.6
− 4.5	0.0007554	0.2403	0.8015	−0.2801	− 4.5
.4	.0007590		.8053	−0.2522	.4
.3	.0007626		.8091	−0.2244	.3
.2	.0007662		.8129	−0.1965	.2
.1	.0007698		.8168	−0.1687	.1
− 4.0	0.0007734	0.2404	0.8206	−0.1408	− 4.0
.9	.0007772		.8246	−0.1128	.9
.8	.0007809		.8286	−0.0848	.8
.7	.0007847		.8326	−0.0567	.7
.6	.0007885		.8366	−0.0287	.6
− 3.5	0.0007923	0.2404	0.8406	−0.0007	− 3.5
.4	.0007960		.8446	+0.0273	.4
.3	.0007998		.8486	0.0553	.3
.2	.0008036		.8526	0.0834	.2
.1	.0008073		.8566	0.1114	.1

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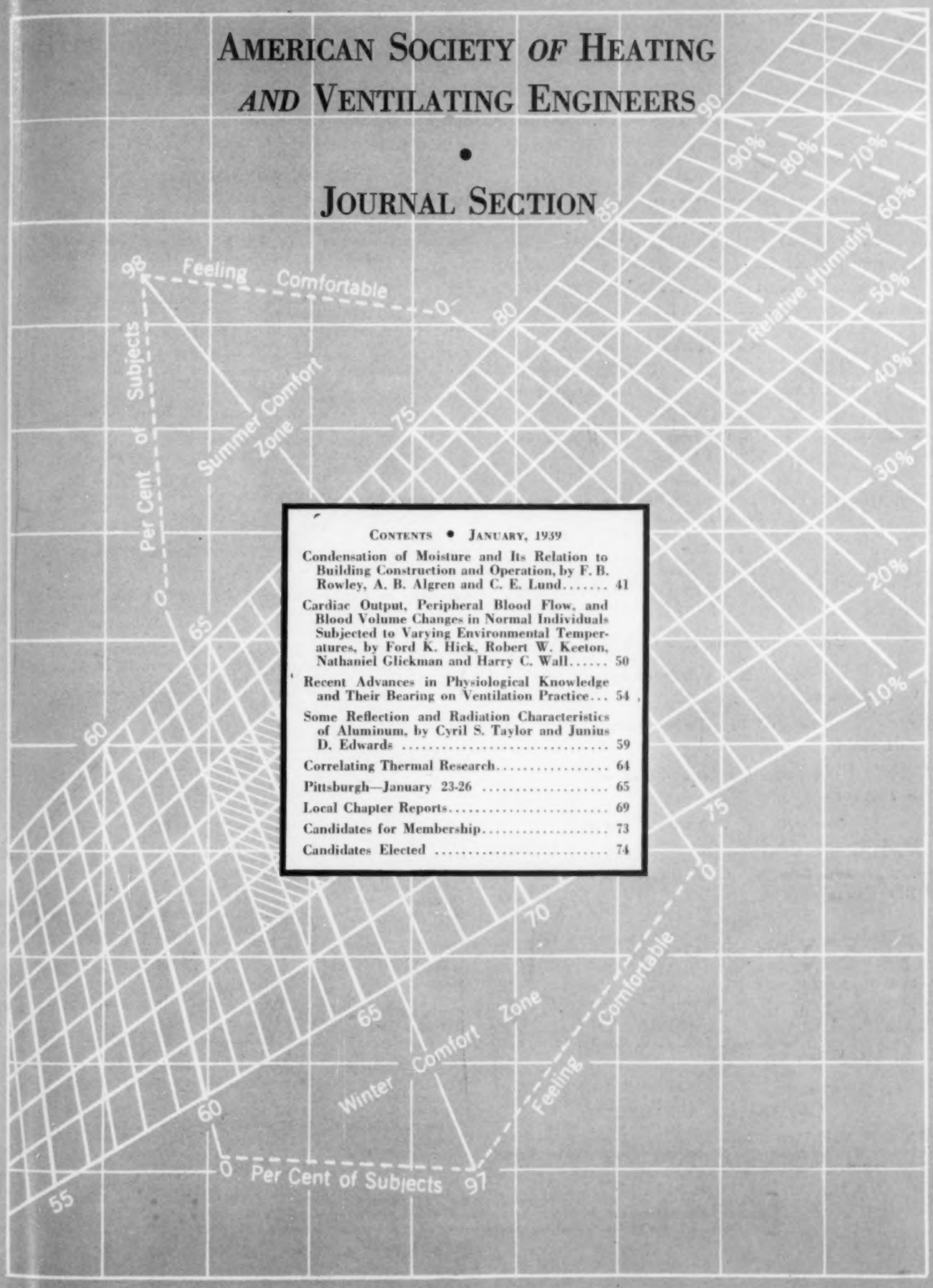

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TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* H. W. SKINNER, 4816 Dexter St., Fort Worth, Tex. *Secretary,* W. H. BADGETT, Texas Engrg. Experiment Station, College Station, Tex.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* S. P. EAGLETON, 3522 "S" St., N. W. *Secretary,* E. V. FINERAN, 411 Tenth St., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* D. W. NELSON, University of Wisconsin, Madison, Wis. *Secretary,* T. M. HUGHEY, 906 N. Fourth St.

Condensation of Moisture and Its Relation to Building Construction and Operation

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IF condensation develops in a building it usually comes from the vapor which is mixed with the air on the warm side of the walls. This vapor travels from points of high vapor density to points of lower vapor density and will pass directly through many types of building construction. The rate at which it may travel is considered to be in direct proportion to the difference in vapor density between two points, and inversely proportional to the vapor resistance of the materials in the path of travel. Vapor density is usually rated in terms of vapor pressure and when the vapor pressure reaches a point which corresponds to the boiling pressure of water at a given temperature, the vapor is saturated and further moisture in the space must exist as mist or fog. As is well known, the maximum vapor density, and therefore vapor pressure, for any given space decreases as the temperature is decreased, and if the temperature of the vapor in any space is gradually decreased, ultimately a point will be reached at which the space is saturated and any further decrease in temperature will cause condensation of part of the vapor within the space. (The vapor density within a space is seldom high enough to cause trouble, unless conditions exist which make it possible for the vapor to accumulate in the form of moisture or frost within the structure.)

There are, in general, two methods by which the vapor may be accumulated either as water or frost. First, certain materials of construction are hygroscopic and will absorb a considerable quantity of moisture directly from the vapor in the air. Second, as the vapor comes in contact with surfaces which are below its dew-point temperature, condensation will take place with the accumulation of water or frost, depending upon the temperature of the surfaces. These surfaces may be the inner exposed surfaces of the walls, windows, etc., or they may be interior parts of walls. The greatest difficulties usually occur from long and severe conditions of moisture accumulation in the form of frost or ice followed by a less severe condition in which the frost is melted at a more rapid rate than it can be vaporized and carried out of the wall. There are several causes for the condition and likewise several remedies. Usually the most severe conditions of condensation are due to a combination of causes, and it is often necessary to apply more than one specific remedy.

Briefly the causes of condensation may be summarized as:

1. High vapor density carried on the warm side of the walls.
2. Allowing this vapor to come in contact with wall surfaces or interior parts of the walls whose temperatures are below their dew-point temperatures.
3. By using exterior construction of walls which may prevent vapor from leaving the walls.
4. Exterior air temperatures which will maintain the interior parts of walls below the dew-point temperature of the vapor in contact with these parts for extended periods.

The first cause may be controlled by the conditions under which the building is operated; the second and third causes may be taken care of in the construction of the building; but the fourth cause must be anticipated from an analysis of U. S. Weather Bureau records and met by selecting the proper types of construction and reasonable operating conditions. The remedies are obviously, first, to lower the relative humidities within the building during cold periods, and second, to provide some type of construction which will prevent the moisture from travelling into the structure and which will make it possible for that moisture which does enter to be vented to the outside. Every case that presents a problem requires an analysis to determine its causes and remedies, and it is not logical to hold any one factor, such as conditions carried within the building or a certain type of building construction, as responsible for condensation until after all of the factors involved have been investigated.

An investigation covering the causes of condensation, the extent of the problem, its effect on building construction, and proper remedies to be applied has been in progress at the University of Minnesota, Engineering Experiment Station, for the past two years. A report[1] describing the test set-up and some preliminary results was given at the Annual Meeting of the Society in January 1938. Several phases of the problem have been solved, but as some of them are inter-related the problem requires additional study. This paper, therefore, must be considered as a second progress report.

During the past year the investigation has followed mainly five lines:

1. A further study of vapor barriers.
2. Ventilation of walls through the exterior surfaces.
3. The effect of vapor barriers on the drying of wet plaster.
4. The effect of attic ventilation on the accumulation of moisture and frost within the attic, and upon attic temperatures.
5. The effect of vapor pressure on the rate of vapor travel through materials.

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[1]Condensation Within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund. (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, January 1938.)

Test Set-up

The complete test set-up was described in the paper previously referred to. Briefly, it consists of an insulated test room 30 ft square and 28 ft high, provided with cooling equipment by which the air temperatures can be maintained continuously as low as –25 F. Small test houses are placed within this room and supplied with conditioned air, the temperature and humidity of which can be accurately controlled to match any conditions which may be met in the field. These test houses are provided with means for inspecting the interior of the walls and for measuring the effect of moisture or frost accumulation during periods of test.

The object has been to simulate as nearly as possible conditions which may be met in the field and to study the causes, effects, and corrective methods for moisture and frost conditions within structures. It has been possible to simulate the conditions in so far as the construction of buildings is concerned, and weather conditions have been duplicated with the exception of the effect of sunshine on a building. It is recognized that the periodic heat of the sun on the surface of a building may have a marked effect on the problem. In most cases, however, the effect of the sun's heat is to alleviate the difficulty, and the conditions set up in the test room are therefore more severe than those met in the field. Any remedies which will take care of test room conditions should be more than safe for equal construction and temperature conditions occurring in the field.

Vapor Barriers

Two methods have been used in studying the comparative values of different types of vapor barriers. First, these barriers have been built into the wall and tested under severe temperature and humidity conditions. Aluminum panels or other vapor-proof materials have been placed on the inside surface of the sheathing to form an effective vapor stop at this point and the accumulation of frost on these vapor-stop materials has been measured and taken as the relative rate at which vapor travels through the particular vapor barrier as built into the wall. It is recognized that this method of test gives relative values only and that the vapor or frost accumulated on the aluminum or test panel is usually more than would be accumulated on the inside surface of the sheathing. It does, however, give a practical rating of the barrier and indicates definitely what may be expected of it in field application. Second, special test panels have been constructed in which the barrier under test forms the main vapor resistance between the vapor on the warm side and the vapor stop on the cold side of the barrier. The details of this test apparatus are shown in the photograph of Fig. 1 and the line drawing of Fig. 2. The individual test panels are made up of 2½ in. of insulation lined on the warm side with the barrier to be tested, and on the cold side with a light weight aluminum panel. The barrier and panel are sealed to the test frame to prevent leakage of moisture and are each spaced about ¼ in. from the insulating material. This test panel is then subjected to a definite set of conditions, such as 70 F air at 40 per cent relative humidity on the warm side and –10 F air on the cold side. Since all the vapor is supposedly stopped at the aluminum panel, the relative humidity of the cold air does not enter the problem. The vapor pressure drop may be taken as the difference between the vapor pressure of the 70 F and 40 per cent relative humidity air on the warm side and the vapor pressure corresponding to the temperature of the surface of the aluminum panel next to the insulation. This is probably not strictly equal to the vapor pressure drop across the barrier, but it gives the same vapor pressure difference for all materials tested by this method, and if the joints are properly sealed between the frame and the panels on each side the method should give the proper relative values for the materials tested. In this test procedure the size of each test sample is $12\frac{13}{16}$ in. × $13\frac{3}{4}$ in., and 16 test samples are built into walls of a small test house.

Tables 1 and 2 give the relative values of different types of barriers when tested in combination with a wall, and Table 3 gives the results of certain barriers when tested by the special test apparatus. The paper barriers of Table 1 were placed on the warm side of the insulation between the metal lath and studs. The paint and surface finish barriers of Table 2 were placed on the inside surface of the plaster. In all cases where test walls were used in combination with the barriers these walls were built with 2 x 4 studs spaced 16 in. on center, finished on the inside with metal lath and plaster and on the outside with 8-in. Ponderosa pine shiplap sheathing, asphalt saturated building paper, and redwood siding, with three coats of white paint. The plaster was approximately ½ in. thick consisting of a scratch coat, a brown coat, and a finish coat. The scratch and brown

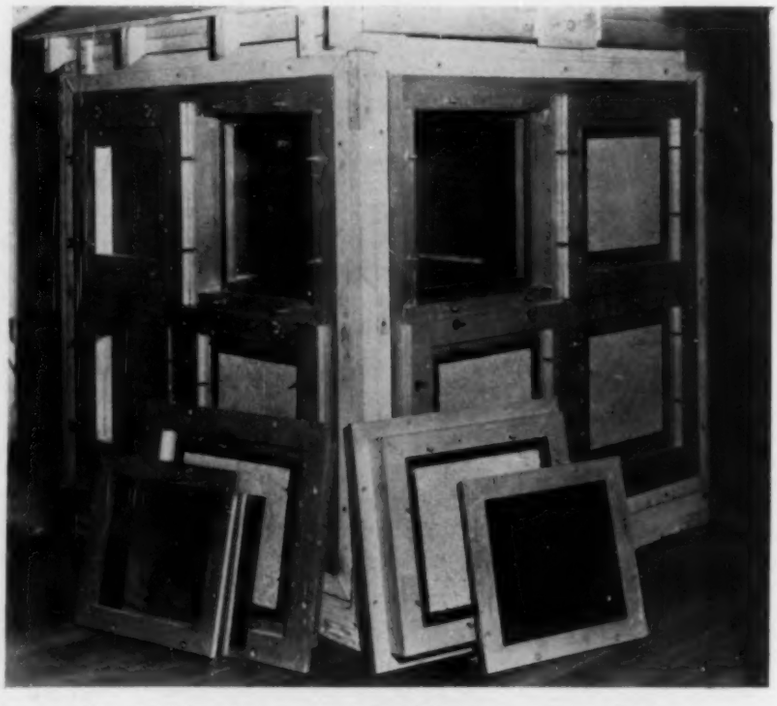

Fig. 1—View of apparatus for making tests on vapor barriers

coats were made by mixing one part of Gypsum plaster and two parts of plaster sand. The finish coat was a thin coat of lime plaster.

Effect of Vapor Barriers on the Drying of Wet Plaster

When an effective vapor barrier is placed underneath plaster the question may arise as to whether or not the plaster will be prevented from drying out by this barrier. If a barrier were 100 per cent effective, all of the moisture from the plaster would necessarily have to pass out to the interior surface, whereas with no barrier it might readily pass out in both directions.

In order to answer this question a test house 4 ft wide, 8 ft long, and 8 ft high was constructed as shown in Fig. 3. The walls were built in sections 4 ft wide and 8 ft high with 2 x 4 studs spaced 16 in. on center; metal lath on the interior surface of the studs; 8-in. Ponderosa pine shiplap sheathing, building paper, and 6-in. redwood lap siding on the outside surface. The sheathing, building paper, and lap siding were built in one unit so that they could be removed as in other standard test panels. Aluminum panels were placed between each stud section next to the sheathing for the purpose of collecting and weighing frost which might accumulate due to the passage of moisture through the wall. An air duct was provided such that cold outside air could be circulated to the interior of the test house at a rate corresponding to two changes per hour for the average house. The walls were constructed with various types of vapor barriers placed under the metal lath, and with 3⅝ in. of Mineral Wool applied between the studs. After the test house was constructed and before any plaster was applied, the inside and outside air temperatures were brought to 70 F and −15 F respectively, and maintained for a sufficient length of time to establish equilibrium through the construction. The plaster was then applied in three coats consisting of a scratch coat, brown coat and finish coat, each allowed to dry for the time indicated. The composition of the plaster was as given in Table 4.

Table 1—Condensation on Inner Surface of Sheathing for Different Types of Vapor Barriers Placed Between Metal Lath and Studs[a]

Vapor Barriers		Outside Air Temp. Deg F	Inside Surface Temp. of Sheathing Deg F	Condensation on Sheathing Grams/Sq Ft/24 Hours		
				Test No.	Test Results	Average of Tests
None	—	−19.5	− 0.3	4	2.16	—
	—	−19.5	− 1.5	4	2.26	—
	—	−19.3	1.3	8	2.02	2.15
None	—	9.9	23.0	7	1.02	—
	—	10.1	22.8	20	1.55	—
	—	9.9	22.5	7	1.65	1.41
Asphalt impregnated and surface coated glossy sheathing paper. Weight of paper tested, 51.9 lb per roll of 500 sq ft	Edges lapped not sealed	−19.5	− 4.1	4	0.09	—
		−19.5	− 2.0	8	0.05	0.07
		9.9	20.5	7	0.00	0.00
	Edges lapped and sealed	−19.5	− 3.5	4	0.02	0.02
		9.9	19.7	7	0.00	0.00
30-30-30 Duplex. Two sheets of 30 lb Kraft paper cemented together with layer of asphalt, equal in weight to 1 layer of paper. Weight of paper tested, 58 lb per roll of 500 sq ft	Edges lapped not sealed	−19.5	− 3.2	4	0.31	—
		−19.3	0.7	8	0.20	0.25
		9.9	20.9	7	0.00	0.00
	Edges lapped and sealed	−19.5	− 3.0	4	0.19	0.19
		9.9	20.7	7	0.00	0.00
Paper with rag felt base saturated with asphalt. Weight of paper tested, 70.8 lb per roll of 500 sq ft	Edges lapped not sealed	−19.5	− 2.2	4	0.52	0.52
		9.9	21.3	7	0.18	0.18
Duplex crepe paper. Weight of paper tested, 40.7 lb per 500 sq ft	Edges lapped and sealed	−19.5	− 1.8	4	0.09	0.09
		9.9	21.6	7	0.00	0.00

[a]Inside air conditions 70 F, 40 per cent R. H.
All walls constructed with 2 in. x 4 in. studs spaced 16 in. O. C.; metal lath and plaster on interior surface; 8 in. Ponderosa pine shiplap, building paper, and 6 in. redwood siding on outside; 3⅝ in. Mineral Wool between studs; vapor barriers as specified.

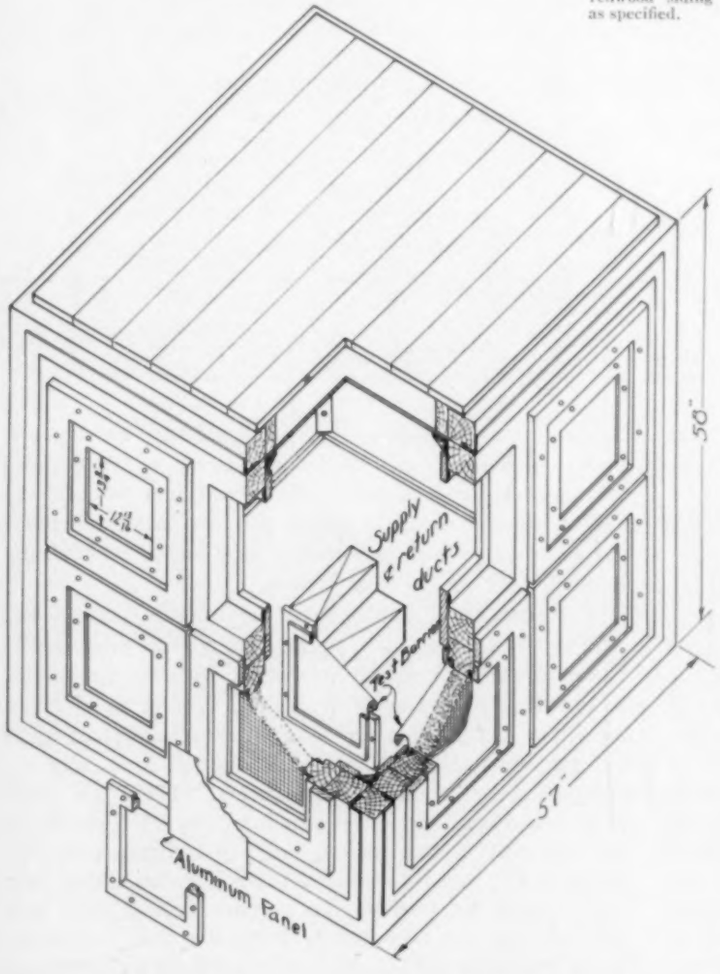

Fig. 2—Line drawing showing details of construction for special vapor barrier test apparatus

The important conclusions which can be drawn from the results of the wet plaster test shown in Table 4 are:

1. A good vapor barrier applied underneath the plaster is effective in stopping the moisture travel from the wet plaster to the interior of the wall, but the relative humidity within the structure may run very high during the drying out period.

2. A barrier should be used which will not be affected by wet plaster.

3. The drying out of the plaster does not appear to be affected by the presence of the barrier and no difference can be detected between the plaster dried with the barrier and that dried without the barrier.

4. During the drying period for plaster over an efficient barrier some moisture may accumulate on the

Table 2—Condensation on Inner Surface of Sheathing for Different Types of Finishes on Interior Surfaces of Plaster[a]

Finishes on Inside Surface of Plaster	No. of Tests	Outside Air Temperature Deg F	Inside Surface Temperature of Sheathing Deg F	Condensation on Sheathing Grams/sq ft/24 Hours
Unfinished	3	—19.5	— 0.2	2.15
	3	+10.0	22.8	1.40
2 Coats Seal Coat Paint[c]	1	—19.5	0.8	0.20
	1	+10.0	23.6	0.00
2 Coats White Flat Paint[b]	1	—19.5	— 1.5	0.24
	1	+10.0	20.6	0.00
2 Coats Aluminum Paint[d]	1	—19.5	0.5	0.25
	1	+10.0	23.6	0.00
1 Coat Asphalt applied hot	1	—19.5	1.1	0.13
	1	+10.0	23.3	0.00
1 Coat Seal Coat Paint[c] 2 Coats White Flat Paint[b]	1	—19.5	1.8	0.23
	1	+10.0	22.5	0.00
1 Coat Seal Coat Paint[c] 2 Coats Aluminum Paint[d]	1	—19.5	3.9	0.16
	1	+10.0	24.3	0.00
1 Coat Glue Size[e] with Plain Wall Paper	1	—19.5	— 0.9	2.11
	1	+10.0	22.2	1.11
1 Coat Glue Size with Dull Surface Treated Canvas Wall Covering	1	—19.5	0.4	0.64
	1	+10.0	22.3	0.10
1 Coat Glue Size with Glossy Surface Treated Canvas Wall Covering	1	—19.5	0.7	0.47
	1	+10.0	21.7	0.01
1 Coat Glue Size with Duplex Crepe Paper[e]	1	—19.5	— 0.2	0.32
	1	+10.0	21.8	0.0

[a]Inside air conditions 70 F and 40 per cent R. H.

All walls constructed with 2 in. x 4 in. studs spaced 16 in. on centers; metal lath and plaster on interior surface; 8 in. Ponderosa pine shiplap, building paper and 6 in. redwood siding on outside; 3½ in. Mineral Wool between studs; surface finish as specified.

[b]*White Flat Interior Paint Specifications:*
Pigment ..... 62.3 per cent
Vehicle ..... 37.7 per cent
Pigment Composition:
Titanium calcium pigment 78.0 per cent
Calcium carbonate ..... 22.0 per cent
Composition of titanium calcium pigment:
Titanium oxide ..... 30.0 per cent
Calcium sulphate ..... 70.0 per cent
Vehicle Composition:
Vegetable oil and resin.. 34.3 per cent
Volatile thinner and drier 65.7 per cent

[c]*Seal Coat Specifications for Priming Coat on Plaster Surfaces:*
Pigment ..... 40 per cent
Vehicle ..... 60 per cent
Pigment Composition:
Titanium calcium pigment .....100.0 per cent
Composition of titanium calcium pigment:
Titanium oxide ..... 30.0 per cent
Calcium sulphate ..... 70.0 per cent
Vehicle Composition:
Resins ..... 16.7 per cent
Vegetable oil ..... 28.8 per cent
Drier and thinner..... 54.5 per cent

[d]*Aluminum Paint Specifications:*
Aluminum paste ..... 1.75 lb
Vehicle ..... 7.40 lb or 1 gal
Composition of paste:
Pure aluminum powder.. 65.0 per cent
Volatile liquid ..... 35.0 per cent
Vehicle:
Water resisting spar varnish.

[e]*Glue Size Specifications:*
A cold water size containing animal glue.
Mixture: 11 pints of water to 1 lb of size.

Note: *Outside White Lead Paint Specifications:*
Pigment by weight..... 70.0 per cent
Vehicle by weight..... 30.0 per cent
Composition of pigment:
A pure white lead paste.
Composition of vehicle:
Linseed oil ..... 90.0 per cent
Drier ..... 10.0 per cent

Table 3—Average Results for Vapor Barrier Tested by Special Test Method[a]

Description of Barrier	No. of Tests	Condensation Grams/sq ft/24 Hours		
		Maximum	Minimum	Average
No barrier	1	—	—	13.40
Asphalt impregnated and surface coated glossy sheathing paper. Weight of paper tested, 51.9 lb per roll of 500 sq ft	5	0.22	0.09	0.16
30-30-30 Duplex Two sheets of 30 lb Kraft paper cemented together with a layer of asphalt, equal in weight to 1 layer of paper. Weight of paper tested, 58 lb per roll of 500 sq ft	3	0.46	0.20	0.37
	2	0.40	0.35	0.38
	2	0.44	0.43	0.43
Paper with rag felt base saturated with asphalt. Weight of paper tested, 70.8 lb per roll of 500 sq ft	3	2.12	1.04	1.46
Single sheet 50 lb Kraft paper, saturated with asphalt, finished weight 68 lb. Weight of paper tested, 12 lb per 500 sq ft	1	—	—	1.75
Single sheet 70 lb Kraft paper, saturated with asphalt, finished weight 82.5 lb. Weight of paper tested, 14.8 lb per 500 sq ft	1	—	—	3.00
Duplex crepe paper, weight of paper tested, 40.7 lb per 500 sq ft	3	0.24	0.11	0.16

[a]Inside air conditions 70 F, 40 per cent R. H.
Outside air conditions—10 F.

Table 4—Air Temperatures and Humidity Conditions for Wet Plaster Tests

Plaster	No. of Hours	Cold Room Deg F	Test House			
			D. B., Deg F	Per Cent R. H.		
				Max.	Min.	Avg.
Scratch Coat[a].....	66	—15.2	69.9	73.0	57.0	63.6
Brown Coat[b].....	96	—15.4	70.1	78.0	59.0	62.7
Finish Coat[c].....	171	—14.3	70.0	75.0	34.0	58.0

[a]Scratch Coat—sand passing No. 10 sieve, 500 lb; fiber plaster, 204 lb; and water, 106 lb.

[b]Brown Coat—sand passing No. 10 sieve, 880 lb; fiber plaster, 353 lb; and water, 185 lb.

[c]Finish Coat—hydrated lime and unfibered plaster to ratio of approximately 8 parts of lime to 1 part of plaster mixed with water to give proper workability.

Note: The mixtures for the scratch and brown coats consisted of 2½ parts of sand by weight, 1 part of plaster, and 6 gal of water per sack of plaster.

warm surface of the barrier, but this will re-evaporate and vent out through the dried plaster without causing any damage to the structure.

Ventilation of Walls to Exterior Air

Vapor passes through all parts of a wall from points of high vapor pressure to points of low vapor pressure. It appears to travel in direct proportion to the differences in vapor pressure, and in inverse proportion to the vapor resisting property of that part of the wall through which it passes. If the conditions of vapor pressure and vapor resistance throughout a wall are such that there is a uniform and equal flow of vapor through each section of the wall, a normal moisture gradient will be established throughout the wall and there will be no point within the wall at which moisture or frost will accumulate. If, on the other hand, conditions permit the vapor to travel at a higher rate through the first section than through the other sections of the wall, there will necessarily be an accumulation of moisture or frost at the line where this rate of travel changes. In order to avoid condensation difficulties it is desirable to construct a wall, which under the conditions of use will resist the entrance of vapor through the warm interior surface, and as a factor of safety permit the free passage through the exterior part of the wall of any vapor which may have entered the interior section.

The passage of vapor from the interior part of the wall to the exterior cold air is often spoken of as ventilation of the wall, and is connected with the movement of air through the exterior surface of the wall. As a matter of fact the venting of vapor may take place without any air movement at all, although it is usually accompanied by some exchange of air. There are cases in practical construction in which it appears desirable to accelerate the normal passage of vapor through the exterior of the wall by means of artificial ventilation. Any exchange of air through the exterior surface of a wall will increase the heat loss through the wall, but under some conditions it is possible to provide enough artificial ventilation to relieve any moisture accumulation within the wall and yet not materially increase the heat loss

through the wall. It is this type of ventilation that is now under investigation. Different methods of ventilating through the exterior surfaces of walls have been applied to both 4-ft and 8-ft walls with different kinds of construction. These tests are still in progress but from the work thus far done it may be said that ventilation of walls to the exterior cold air will increase the rate of vapor travel through the exterior surface and give partial or complete relief from any condensation difficulties depending upon the severity of conditions. For severe conditions it usually will be necessary to consider additional corrective measures such as reducing the inside relative humidity or using some vapor resistant material on the warm side of the wall.

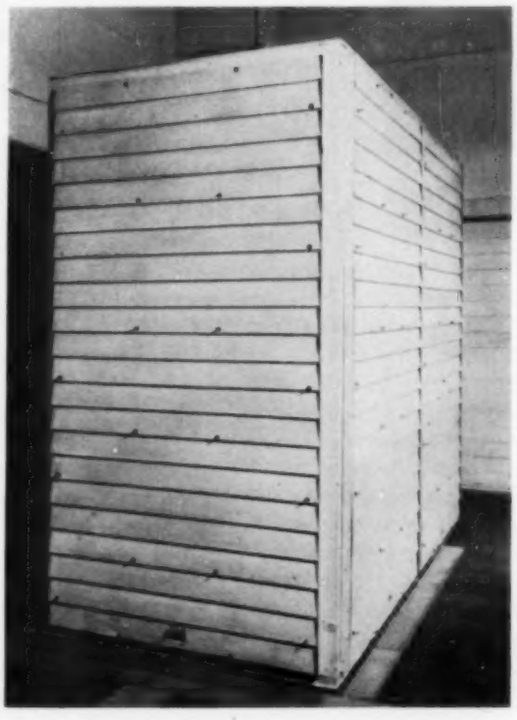

Fig. 3—Test house used for wet plaster test and also subsequent test on wall ventilation

Ventilation of Attic Space

When insulation is applied to the ceiling of the upper floor and the attic space above is unheated, the under side of the roof and attic walls may become very cold and provide a surface for the condensation of moisture and accumulation of frost, if vapor is allowed to pass through the upper floor ceiling into the attic space. The most effective way of preventing this condensation is to place a vapor barrier on the warm side of the insulation between the attic space and the rooms below. In case it is not practical to use this type of a barrier, or as a precaution against any moisture which may pass the barrier, it is usually possible and practical to ventilate the attic space to the exterior of the building. The question then arises as to the type of ventilation system to use and the minimum amount of air necessary for effective ventilation.

In the preliminary investigation of the possibilities of attic ventilation three of the small size test houses have been equipped with attics. The ceilings below were constructed with 2 x 4 joists, metal lath and plaster, with 3⅝-in. Mineral Wool between the joists. No vapor barriers were used. The roofs were constructed with 8-in. pine shiplap sheathing covered with roofing paper. The attic spaces were typical for cold attics with cold surfaces underneath the roofs for the condensation of moisture. The first attic was completely enclosed without ventilation; the second was provided with adjustable vent openings in each gable; and the third was provided with mechanical ventilation by which the amount of air supplied to the attic could be accurately measured. In each of the attics the roof boards with attached roofing material were built as a unit so that the entire roof section could be removed during test for inspection of interior conditions if desired. Small condensation panels were also placed on the under side of the roof sections to provide means of collecting and weighing the frost accumulated during a test period. Observation openings were provided in the gables for inspection of the interior to determine whether or not condensation was taking place during a test period without disassembling the walls. Fig. 4 shows a line drawing of the attic in which openings were provided for natural ventilation through the gables. Fig. 5 shows the three test attics assembled in the cold room together with the pump and meter used to furnish air to the mechanically ventilated attic.

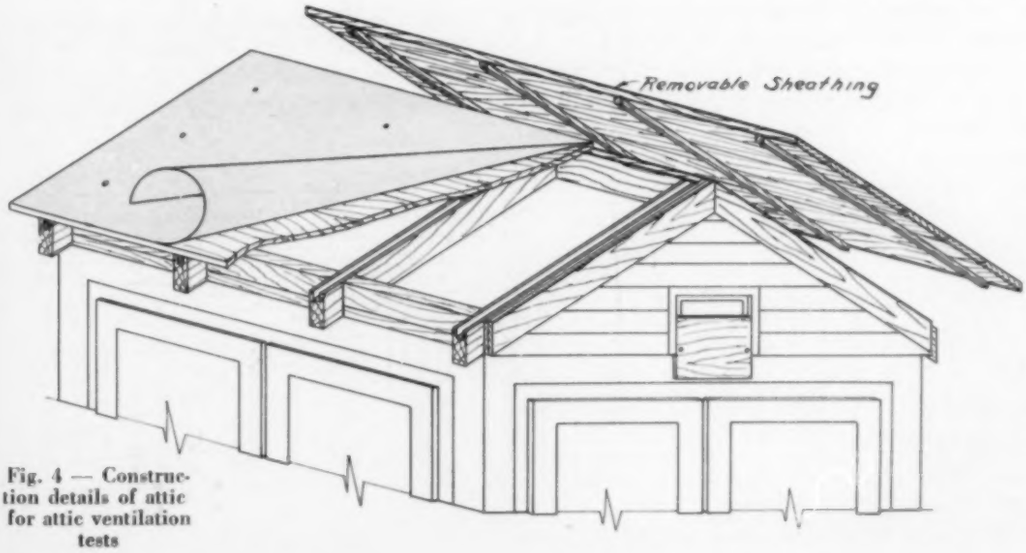

Fig. 4 — Construction details of attic for attic ventilation tests

The test data thus far taken on the three attics are shown in Table 5. Tests have been run with outside temperatures ranging from −10 F to +25 F and with air conditions

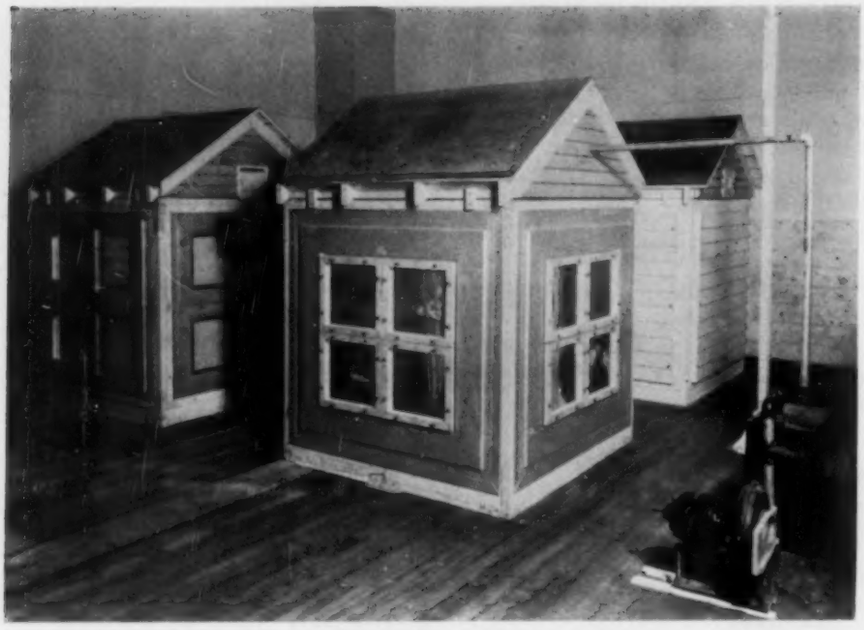

Fig. 5—View of set-up for ventilated and unventilated attics. Apparatus in foreground used for measuring supply of air to mechanically ventilated attics

in the test houses ranging from 70 F and 40 per cent relative humidity to 70 F and 20 per cent relative humidity.

From the results in Table 5, test 19, period 1, which was run at a rather severe condition of 70.3 F, 40.5 per cent relative humidity for inside air, and −10.4 F for outside air, a rate of condensation on the underside of the roof for the unventilated attic of 3.01 g (grams) is shown. Under the same conditions the attic with natural ventilation having a ¼ sq in. opening on each gable per square foot of ceiling area, and the attic with mechanical ventilation having 3 cu ft of air per hour per square foot of ceiling area, showed no condensation. Test 19, period 2-3 run under the same temperature and humidity conditions gave 3.33 g condensation for the unventilated attic and 0.16 g for the attic with natural ventilation with opening at each end reduced to ⅛ sq in. per square foot of ceiling area, and 0.03 g for the mechanically ventilated attic supplied with 1½ cu ft of air per square foot of ceiling area per hour. Evidently test period number 2-3 represents the minimum of ventilation for either natural or mechanical ventilated attic.

The loss of heat due to attic ventilation may be calculated for both the natural and mechanical ventilation by assuming that the loss of heat through each ceiling is directly proportional to the drop in temperature from inside air to attic air, and that all ceilings have the same conductance values. On this basis the increase in heat loss through the ceilings during period number 2-3 for the natural ventilated attic as compared with that of no ventilation is 5 per cent, and for the mechanically ventilated attic it is 4 per cent. An average of all the tests thus far run seems to indicate a heat loss due to ventilation of approximately 5 per cent for the conditions given in test period number 2-3.

The second approximate method of figuring the heat loss due to mechanical ventilation would be to assume that the loss is equal to the amount of heat required to heat the ventilation air from outside temperature to attic temperature. On this basis the heat loss caused by mechanical ventilation for period number 2-3 would be the amount required to heat 1.5 cu ft per hour through a temperature range from −10.3 F to +2 F, or 0.39 Btu per square foot of ceiling area per hour. If a coefficient of heat transmission for the complete ceiling of 0.08 and a temperature drop of 65.4 are assumed as in the unventilated attic the heat loss per square foot of ceiling area for the unventilated attic is 5.23. Thus the calculated loss would be 7.4 per cent. This method would give an excessive loss charged against ventilation since it assumes that there is no loss for the unventilated attic. It seems safe to assume that the loss due to adequate ventilation would not be more than 10 per cent when based on the conductivity through the ceiling, and excepting for the most severe conditions it should be less than this amount. The data thus far obtained have been on small sized attics and of a single type of roof. The application to larger sized attics will be investigated later.

Relation of Vapor Conductance Through a Material to the Drop in Vapor Pressure

There has been much speculation as to whether the rate of vapor travel through a material or combination

Table 5—Attic Ventilation

Test No.	Period No.	Duration of Test	Outside Air Temp. Deg F	Inside Air Conditions		No Ventilation		Natural Ventilation			Mechanical Ventilation		
				Deg F	R. H. %	Attic Air Temp.	Condensation[a]	Opening sq in.[b]	Attic Air Temp.	Condensation	Air Supplied cu ft[c]	Attic Air Temp.	Condensation
19	1	54	−10.4	70.3	40.5	4.6	3.01	0.25	− 0.4	0.0	3.0	1.8	0.00
19	2-3	68	−10.3	70.0	40.9	4.6	3.33	0.125	1.2	0.16	1.5	2.0	0.03
20	1	66	+15.0	70.1	40.6	26.9	0.0	0.125	24.2	0.0	1.5	24.3	0.00
20	2	99	+10.0	69.9	40.3	22.6	0.0	0.125	19.9	0.0	1.5	19.7	0.00
20	3	117	+ 5.0	70.1	40.6	17.3	1.16	0.125	13.9	0.0	1.5	14.4	0.00
20	4	72	+ 5.0	70.0	31.3	17.6	0.78	0.125	14.6	0.0	1.5	15.0	0.00
20	5	72	+ 5.0	70.1	20.7	16.9	−1.02	0.125	14.1	0.0	1.5	14.4	0.00
20	6	98	+10.0	70.1	20.6	20.5	−1.22	0.125	18.3	0.0	1.5	18.4	0.00
20	7	66	+15.0	70.1	21.0	23.4	0.0	0.125	21.2	0.0	1.5	21.2	0.00
20	12	141	0.0	70.0	21.3	11.8	1.76	0.125	7.8	0.0	1.5	9.6	0.00
20	13	72	−11.4	70.1	19.9	1.43	2.52	0.125	− 2.5	0.0	1.5	− 0.5	0.00

[a]Condensation in grams per square foot ceiling area per 24 hours.
[b]Opening in square inches per square foot ceiling area in each gable.
[c]Air supplied in cubic feet per hour per square foot ceiling area.

of materials as in a wall is in direct proportion to the drop in vapor pressure along the path of vapor flow, or whether the rate follows some other law. It is probably true that the manner in which the vapor is conducted through various parts of the path for various combinations of material is not the same for all conditions of vapor flow. For certain cases such as impervious hygroscopic materials the vapor may be hygroscopically absorbed into one surface of the material and transmitted through to the opposite side, as a liquid saturating the material, from where it will be re-evaporated. In other cases it may be transmitted directly through the material as a vapor without any change in state. The laws governing the rate of transfer may be affected by temperature range, method of transfer, or by some property of the material. If, however, the rate of transfer is substantially in direct proportion to the vapor pressure drop along the path of transfer, then the problem of vapor flow may be considered similar to that of heat flow through materials, and vapor conductance coefficients may be developed for individual materials and combinations of materials similar to the heat conductance coefficients now in use.

Sufficient test data have not yet been taken to prove whether or not the rate of vapor flow through the frame walls now being investigated may be considered substantially proportional to the vapor pressure drop. In order to get some indication in regard to this law, test values for several walls were selected and compiled in Table 6. These test values are either from identical walls or from walls constructed from the same specifications. They were all frame walls with metal lath and ½-in. Gypsum plaster on the inside surface, and pine sheathing, building paper, and lap siding on the outside surface with 3⅝ in. Mineral Wool between the studs. They were all tested at substantially 70 F inside air temperature with humidities of approximately 20 per cent and 40 per cent for outside temperatures of 5 F, 10 F, 15 F, and –11.4 F. The test results for these walls are arranged in the table to show the comparative values for humidities of 20 and 40 per cent at the different cold room temperatures.

Since in each pair of tests all test conditions were the same with the exception of the vapor pressure on the warm side of the wall, the rate of vapor flow from the sheathing surface to the outside air should be the same for both tests of a pair, and the difference in frost accumulation on the sheathing should be caused entirely by the difference in rate of vapor flow from warm air to the sheathing. The only change in conditions which could cause a change in rate of flow is in the vapor pressure on the warm side, and it is reasonable to assume that this difference in vapor pressure drop between the two tests of a pair was entirely responsible for the difference in frost accumulation on the sheathing for the two tests.

Consider the first two lines of test results in the table. In this case the same wall was tested at a cold room temperature of 15 F and conditioned air on the inside of the test house of 70 F with 40.7 per cent relative humidity for the first test and 70 F and 20.8 per cent relative humidity for the second test. In this pair of tests the conditions were all substantially the same with the exception of the vapor pressures on the warm side of the test wall. These were 0.3003 in. of mercury for the first test and 0.1543 in. of mercury for the second test, giving a difference of 0.146 in. of mercury. The rate of condensation on the aluminum panel lining the inside of the sheathing was 1.53 g per square foot per 24 hours for the first test and –0.29 g for the second test, giving a total difference of 1.82 g. The rate of condensation per inch of pressure difference is therefore $\frac{1.82}{0.146}$ or 12.46. This value may be considered as the vapor conductance value in grams of moisture per square foot of area per 24 hours per inch of mercury pressure drop for the combination of plaster, metal lath and insulation up to the sheathing, and for convenience may be designated as V_1. A similar calculation for the other pairs of test values shown in the table gives the individual values as listed with an average of 12.19 for the five cases. The variation in these values is not great considering the test method used, and the results indicate that the flow of vapor is in direct proportion to the drop in vapor pressure along the path of flow.

A significant point to be noted from Table 6 is the marked effect of relative humidity of the inside air on the rate of condensation within the wall. For outside temperatures of 10 or more degrees above zero a drop in relative humidity of the inside air from 40 to 20 per cent entirely cleared up any condensation difficulties and reduced the amount of moisture previously accumulated in the wall. At 5 F outside air temperature the rate of condensation was reduced by about 80 per cent, and at –11.4 F it was reduced 71 per cent by reducing the relative humidity of the inside air from 40 to 20 per cent. From these results it is evident that high relative humidity is one of the major factors in condensation problems and one of the first points which should be investigated when condensation occurs. Relative humidities of inside air should be controlled with reference to outside weather conditions.

Table 6—Test Results Indicating the Rate of Vapor Travel Through Material Is Proportional to the Vapor Pressure Difference

Test and Period No.	Outside Air Temp. Deg F	Inside Air Conditions		Vapor Pressure In. Hg			Condensation on Sheathing Grams/Sq Ft/24 Hours	V_1[a]
		Deg F	R. H. %	Inside Air	Inside Surface of Sheathing	Outside Air		
20–1	15.0	70.2	40.7	0.3003	0.1392	0.0565	1.53	.....
20–7	15.1	70.1	20.8	0.1543	0.1392	0.0568	—0.29	12.46
20–2	10.1	69.9	40.6	0.2992	0.1175	0.0442	1.55	.....
20–6	10.0	70.1	20.6	0.1521	0.1175	0.0440	—0.44	13.50
20–3	5.0	70.3	40.7	0.3037	0.0980	0.0342	1.93	.....
20–5	5.0	70.1	20.6	0.1521	0.0980	0.0342	0.40	10.10
20–2	—11.4	70.4	40.0	0.3016	0.0566	0.0144	2.66	.....
20–13	—11.4	70.0	20.0	0.1480	0.0561	0.0144	0.97	11.05
7–4	9.9	70.0	41.0	0.3029	0.1159	0.0438	1.65	.....
20–6	10.0	70.1	20.6	0.1521	0.1157	0.0440	—0.44	13.82

[a] V_1 = Calculated vapor transmittance coefficient from inside air to inside surface of sheathing in grams per square foot per 24 hours per inch of Hg pressure difference.

Effect of Variable Outside Temperature on the Rate of Moisture Condensation in a Wall

In most of the tests thus far made the cold room temperature has been maintained constant throughout an extended test period and the frost accumulation in the different parts of the walls has been due to continuous

uniform temperature difference between the inside and outside air. In practice the outside air temperature does not remain constant throughout long periods, and even for rather extended cold periods there is a variation in outside temperature over the 24 hour period. The daily variation of temperature may be taken as approximately 15 F, but the variation through any ordinary low temperature period will be much greater than this. The question naturally arises as to whether the condensation conditions within a wall are the same when the wall is subjected to a variable outside temperature condition as they are for the same wall subjected to a constant outside temperature condition which would be equal to the average of the variable temperatures, all other conditions remaining the same.

In order to compare the results which might be expected for a given wall operating under uniform outside temperature conditions to those for the same wall operating under variable outside temperatures having the same average as the uniform temperatures, consider a frame wall with 3⅝ in. of insulation between the studs. Consider this wall to be subjected to a test condition consisting of warm air on the inside of the wall of 70 F and 40 per cent relative humidity, and cold air on the outside of 0 F and 50 per cent relative humidity, and compare the probable result for the same wall subjected to the same inside air conditions but with an outside temperature ranging from –20 F and 50 per cent relative humidity to +20 F and 50 per cent relative humidity, thus having an average outside air temperature of zero.

Table 7—Calculated Temperatures and Vapor Pressures Through Insulated Wall for Different Outside Air Temperatures—Other Conditions Remaining Constant

Inside Air		Outside Air		Inside Surface Temp. of Sheathing	Vapor Pressure		
Temp. Deg F	R.H. %	Temp. Deg F	R.H. %		Inside Air	Sheathing	Outside Air
70	40	0	50	11.9	0.2955	0.0693	0.0189
70	40	–20	50	— 4.75	0.2955	0.0321	0.0063
70	40	20	50	28.5	0.2955	0.1535	0.0514

(1) $(0.2955 - 0.0693)\ V_1 - (0.0693 - 0.0189)\ V_2 =$ rate of condensation for zero degrees $0.2262\ V_1 - 0.0504\ V_2 = R_1$.
(2) $(0.2955 - 0.0321)\ V_1 - (0.0321 - 0.0063)\ V_2 =$ rate of condensation for — 20 F $0.2634\ V_1 - 0.0258\ V_2 = R_2$.
(3) $(0.2955 - 0.1535)\ V_1 - (0.1535 - 0.0514)\ V_2 =$ rate of condensation for 20 F $0.1420\ V_1 - 0.1021\ V_2 = R_3$.
(4) Average of (2) and (3) $0.2027\ V_1 - 0.0639\ V_2 =$ rate of condensation for average of two conditions +20 to —20 F.

Table 8—Relation Between Condensation Rate for Constant Outside Temperatures and for Variable Outside Air Temperatures for Walls and Attics

Test and Period No.	Outside Air Temp. Deg F	Inside Air Conditions		Vapor Pressure In. Hg			Condensation Rate on Sheathing Grams/sq ft/24 Hours
		Deg F	R.H. %	Inside Air	Outside Air	Inside Surface of Sheathing	
Walls							
20–6	+10.0	70.1	20.6	0.1521	0.0440	0.1175	—0.44
20–12	0.0	70.1	20.5	0.1514	0.0266	0.0815	0.64
20–13	—10.0	70.0	20.0	0.1480	0.0144	0.0561	0.97
Attics							
20–6	+10.0	70.1	20.6	0.1521	0.0440	—	—1.22
20–12	0.0	70.1	20.5	0.1514	0.0266	—	1.76
20–13	—10.0	70.0	20.0	0.1480	0.0144	—	2.52

Table 7 shows the temperatures and vapor pressures for the air on the warm side, the inner surface of sheathing, and the air on the cold side of the wall for the three test conditions. If it is assumed that the rate of vapor travel between two parts of the wall is equal to a constant multiplied by the drop in vapor pressure between these parts, then equations numbers (1), (2) and (3) show the rates of frost accumulation on the sheathing for the three test conditions, and equation number (4) shows the average for conditions (2) and (3) which is to be compared with test condition (1). An inspection of equations numbers (1) and (4) shows that the rate of vapor travel from the inside air to the sheathing is greater for the constant air temperature condition than it is for the variable air conditions. At the same time the rate of vapor travel from the surface of the sheathing to the outside air will be greater for the condition of variable air temperature than for the condition of constant air temperature.

In order to check the correctness of these calculated results test data have been compiled in Table 8 for three tests, the first with a cold room temperature of +10 F, the second 0 F, and the third –10 F. All other conditions were maintained substantially the same throughout the three tests. In the last column of the table the rates of condensation are shown, first on the surface of sheathing of an insulated wall without vapor barrier, and second on the underside of the roof over an unventilated attic with insulation in ceiling below. The average rate of condensation on the sheathing of the wall for the two tests of +10 F and –10 F is 0.265 g, whereas for the test at zero degrees it is 0.64 g per square foot of sheathing surface per 24 hours. For the underside of the roof over the attics the average rate for the two tests of +10 F and –10 F is 0.65 g as compared with 1.76 g per square foot of ceiling area at the zero degrees temperature. These test results bear out the calculated results as taken from figures of Table 7. It should be noted that the decrease in rate of condensation from –10 F to +10 F outside temperature is greater for the roof over the attic than it is for the wall. This is due to the fact that for the attic the area through which the condensation leaves the attic is equal to the roof and gables, or about double the ceiling area, whereas in the wall the sheathing area is only equal to the inner surface area. Thus when the rate of vapor travel from the condensation area is increased it has a greater effect on the rate of accumulation for the attic than it would have for the wall.

These test results demonstrate the theory shown in the calculations and indicate that the test conditions of constant outside air temperatures are more severe in so far as condensation troubles are concerned than equivalent variable air temperatures would be.

Conclusions

The conclusions reached from the investigation thus far made are as follows:

1. There are several fundamental causes for condensation within buildings, and in general they may be classified under three headings:
   a. Operating conditions for buildings.
   b. Construction of buildings.

c. Outside weather conditions.

In making an analysis of any problem all three should be considered, and in making corrections the causes under (*a*) and (*b*) must be coordinated to meet the requirements under (*c*).

2. The operating conditions which are usually subject to control are:

a. Relative humidity carried within buildings.

b. Controlled ventilation of structures.

3. The overall vapor transmittance for a wall or combination of materials appears to be directly proportional to the vapor pressure drop across the wall providing that there is no condensation. The accumulation of moisture within any section of a wall is equal to the vapor transmittance from inside air to that section minus the vapor transmittance from that section to outside air. Reducing the relative humidity of the air within a building will reduce the rate of moisture condensation within the structure by a much greater percentage than the percentage of change in the humidity.

4. It is possible to reduce the rate of condensation within the structure by ventilating to the outside. This method may be particularly effective in attics where the condensation occurs on the underside of the roof. Adequate ventilation may be obtained without serious loss of heat.

5. Those factors which should be considered in new construction are:

a. Build with special reference to vapor-proof construction on the inside of a building to prevent moisture from entering the walls or interior structure.

b. Build with specific reference to allowing the vapor to vent through outside walls either with or without interchange of air.

c. Use sufficient insulation to keep all interior surfaces above the dew-point temperature of the inside air.

6. A good vapor barrier on the warm side of insulation is an effective method of dealing with the condensation problem at the time of construction.

7. A barrier underneath the plaster will not interfere with the drying out of the wet plaster or affect the quality of the finished plaster.

8. A good vapor barrier construction may be obtained with a vapor-proof paper properly applied under the plaster or a vapor-proof finish on the interior surface of the wall.

9. For cold attic spaces it is desirable to allow openings for outside air circulation through attic space as a precaution against condensation on the underside of the roof even though barriers are used in the ceiling below.

Acknowledgments

This paper gives a part of the results which have been obtained during the past year in a cooperative research program between the *National Mineral Wool Association* and the Engineering Experiment Station, University of Minnesota. The purpose of the program has been primarily to determine the cause of condensation within buildings, its probable extent under various conditions, and the best methods of preventing and correcting the same.

The authors wish to acknowledge the cooperation of the *National Mineral Wool Association* and the assistance of the Technical Committee of the Association in the entire program. This committee consisting of J. S. Irvine, *chairman;* W. W. Cullin, E. W. McMullen, E. R. Williams, and C. L. Neumeister has contributed liberally to the success of the project.

New Book on Forced Circulation Heating

In a sizeable volume entitled Winter Air Conditioning: Forced Air Heating, by S. Konzo, Research Assistant Professor in the Department of Engineering of the University of Illinois, both scientific and practical information on forced air heating has been compiled for architects, builders, heating and ventilating engineers, manufacturers of heating equipment and heating contractors. The book embodies the detailed results of the exhaustive tests of forced air heating and winter air conditioning which have been going on for many years in the Research Residence at the University of Illinois, in connection with the research project financed by the *National Warm Air Heating and Air Conditioning Association.* As a practical manual on the design, construction, installation and actual operation in the home of the modern forced air heating system, it is an important contribution to the extensive literature on heating and ventilating.

The book includes not only the complete University of Illinois test data to date, but brings together a considerable amount of other material not previously available in one volume; all of it bearing directly on one or another practical aspect of the house-heating job. Other authorities are quoted freely, and due credit is given in every instance to their contributions to modern efficiency, economy and home comfort.

The main job, however, is of course Professor Konzo's. He has done an excellent piece of work, not only in bringing together and arranging in clear and orderly fashion such an immense amount of diverse and useful information, but in presenting simply, logically and understandably, the information gained from the extremely wide and varied range of tests with which he and his associates have been from the beginning identified.

Quotation of a few of the 32 chapter headings will show the scope of the book and the completeness with which it covers everything the architect, the builder, the manufacturer or the heating contractor needs to know about modern forced air heating. These headings include: Comparison of the Gravity Warm Air Heating Plant and the Forced Air Heating Plant; Winter Air Conditioning and Human Comfort; Comfort Chart and Effective Temperatures; Humidification Requirements and Limitations; Humidification and Humidity Controls; Filtration of Air and Air Filters; Heat Loss Calculations for Heating; The Use of Dampers in Duct Systems; Rational Approach to Design of Duct System; Selection of Furnaces and Burners; and Chimneys and Drafts.

The book is $3.00 per copy and may be obtained from the *National Warm Air Heating and Air Conditioning Association,* 11 West 42nd Street, New York, or 50 West Broad Street, Columbus, Ohio.

Cardiac Output, Peripheral Blood Flow, and Blood Volume Changes in Normal Individuals Subjected to Varying Environmental Temperatures

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Department of Medicine of the University of Illinois

THE human body maintains a nearly constant temperature by balancing its heat production against heat loss. DuBois has pointed out that in a man[1] millions of calories of heat produced are exactly balanced by heat loss over a period of years with surprising facility. In the *comfort zone*[2] a human maintains normal temperature and a warm skin while losing 25 per cent of his heat by evaporation.[3] In the *zone of cooling*[4] a man loses heat largely through radiation until he chills, begins to shake, and raises his heat production to recover his normal temperature. In the *zone of evaporative regulation*,[5] he loses more than 25 per cent of his heat through the evaporation of water, and maintains a normal body temperature until this mechanism fails. The whole study of heat regulation is then one of the physiological adjustments of the body to meet a physical problem in thermodynamics. When these adjustments occur smoothly and escape the attention of the individual he is *comfortable*. When they proceed with more difficulty he is *uncomfortable*. Comfort then becomes an index of the easy adaptation of the heat loss mechanism to the needs of the moment.

In studies by DuBois, Winslow, and associates, many facts relative to heat loss through the skin have been detailed. They have noted that the delivery of heat from the deeper parts of the body to the surface is accomplished chiefly by the circulation of the blood. It, therefore, seemed imperative to make a quantitative study of the circulatory adjustments concerned in the transport and dissipation of heat. The present report details observations on the measurements of cardiac output, the quantity of blood flowing through the periphery, and the circulating blood volume. It is obvious that man must be used as the experimental subject in the physiology of heat loss, since he differs so widely from animals.

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Plan of Experiment

Healthy subjects were studied under basal conditions in an atmosphere within the range of the *comfort zone*. When subjects have fasted for 12 hours and have remained quiet for one to two hours after a good night's rest they are under basal conditions. These subjects have a constant heat production varying little from one day to another.

The atmosphere was next changed to the *zone of evaporative regulation* so that all the heat was lost by evaporation. The humidity governs the ease of heat loss in this evaporative zone, so the subjects were submitted to hot dry and hot wet conditions. All measurements were carried out two to three times on adult, normal subjects, one male and one female.

The general procedure was such that the subject was in the hospital over-night, rested in the air conditioned room if it was comfortable, and remained quiet in the early morning until the work for that day was completed. If the room was uncomfortable, the subject slept in another room in the hospital and was wheeled into the air conditioned room between 5 and 6 A. M. and observations began two hours or more afterward. All observations were therefore under post-absorptive or *basal* conditions and with a nude male subject and a practically nude female subject. The subject was on a bed.

Procedure

The man's age was 22, height 74¾ in., weight 168 lb, DuBois surface area 2.0 square meters. The woman's age was 24, height 63 in., weight 105 lb, DuBois surface area 1.45 square meters.

Observations were made at three general conditions: a comfortable condition (about 84 F dry-bulb and 60 F

Table 1—Cardiac Output

Subject	Condition	Dry Bulb, Temperature, F	Wet Bulb, Temperature, F	Oxygen Consumption, cc/min	Arterio-Venous Oxygen Difference	Cardiac Output, L/Min	Rectal Temperature, F	Pulse
J. Morgan (Female)	Comfortable	84.0	63.5	188.5–187.5	69.5	2.7	.....	64
		84.0	64.6	185.0–178.5	77.7	2.4	98.4	63
		84.0	61.7	178.3–190.5	64.9–76.5	2.7-2.5	97.9	63
		Average		184.7	72.1	2.6	98.2	63
	Hot and Dry	99.4	65.0	185.5–188.3	61.3–73.4	3.0–2.6	98.8	70
		99.4	67.5	181.3–183.0	66.4–73.5	2.7–2.5	98.4	74
		99.5	71.6	185.8–182.2	62.0	2.9	98.4	68
		Average		184.4	67.3	2.7	98.5	71
	Hot and Wet (No fever)	98.7	84.5	176.7	66.0	2.7	98.8	74
	Hot and Wet (Fever)	99.0	87.2	185.0	52.0–57.1	3.5–3.2	99.2	74
		98.8	86.1	193.3	52.8	3.7	99.2	75
		99.5	90.0	215.3	51.1–55.9	4.2–3.9	99.5	87
		Average		197.9	53.8	3.7	99.3	79
T. Schertz (Male)	Comfortable	87.0	65.8	286	70.5–65.0	4.1–4.4	97.9	55
		85.8	63.8	268–252	60.3–65.3	4.3–4.0	97.4	48
		Average		268.6	65.2	4.2	97.7	52
	Hot and Dry	101.0	75.5	261.5–280	56.0–55.5	4.8–4.9	99.1	49
		99.5	64.5	254.0–245.3	54.3	4.7	98.5	47
		Average		260.2	55.3	4.8	98.8	48
	Hot and Wet (Fever)	100.0	83.0	285.0–287.0	67.1–68.3	4.6–4.2	99.0	56
		99.3	86.0	248.2–266.2	39.4	6.3–6.8	99.4	68
		101.0	90.0	345	50.7–55.5	6.8–6.2	101.4	80

wet-bulb), a hot dry, and a hot wet condition. To insure that all the heat loss would be by evaporation in the hot conditions, the dry-bulb was set at 99.5 F and the wet-bulb adjusted to between 65 to 74 F for dry, and 85 to 91 F for wet conditions. The two hot periods differ only in moisture content. Air movement was kept constant and minimal. No drafts which would activate a precision balanced vane type anemometer were detected. The temperature of the walls was uniformly equivalent to air dry-bulb temperature.

Cardiac output was measured by the method of Grollman[6] as modified by McMichael[7] to use the Van Slyke gas analytic apparatus. The oxygen consumption of the subjects was determined by analysis of expired air collected in a spirometer (Tissot tank). Skin temperature was obtained by a multiple lead constantan-copper thermocouple apparatus patterned directly after that of Sheard[8], using a potentiometer for measurement of electro-motive force (E. M. F.). The mean skin temperature, an average figure weighted according to the surface area of the parts studied, was calculated in the manner of Hardy and DuBois,[9] after measurement of 16 separate points. Blood volume studies were carried out using a technique described by Gibson and Evelyn[10] with surprising ease in reproducing results.

In determining venous oxygen saturation blood was drawn from a superficial vein at the elbow without a tourniquet. The samples were collected without contact with air and analyses were carried out as described by Van Slyke et al.[11]

Measurement of the rate of insensible loss of weight was carried out on a balance sensitive to less than one gram.

[6]The Cardiac Output of Man in Health and Disease, by Arthur Grollman. (Thomas, Springfield, Ill., 1932.)

[7]Postural Changes in Cardiac Output and Respiration in Man, by J. McMichael and F. A. Johnston. (*Quarterly Journal of Experimental Physiology*, 27:55, 1937.)

[8]The Electromotive Thermometer: An Instrument and a Method for Measuring Intravenous, Superficial, and Cavity Temperatures, by Charles Sheard. (*American Journal Clinical Pathology*, 1:209, 1931.)

[9]The Technic of Measuring Radiation and Convection, by J. D. Hardy and E. F. DuBois. (*Journal of Nutrition*, 15:461, 1938.)

[10]Clinical Studies of the Blood Volume IV. Adaptation of the Method to the Photoelectric Microcolorimeter, by J. G. Gibson, II and K. A. Evelyn. (*Journal of Clinical Investigation*, 17, 153, 1938.)

[11]Quantitative Clinical Chemistry, by J. P. Peters and D. D. Van Slyke. (Vol. II, p. 337, Williams and Wilkins Co., Baltimore, 1932.)

Results

Cardiac Output

Table 1 lists the essential data on cardiac output on these subjects. The rectal temperatures show anticipated variations. Subjectively the hot, wet atmosphere was extremely uncomfortable. In this condition the elimination of heat is difficult and at times there is an actual retention. This is shown by the elevation of the rectal temperature above 98.6 F. With a retention of heat there is an elevation of the oxygen consumption and a corresponding increase in cardiac output. Unless such a retention of heat occurred there was no change in the cardiac output. The pulse rate is very low under comfortable conditions on these two subjects. It is somewhat higher in the hot dry conditions in the woman. Obviously the pulse rate changes do not parallel changes in cardiac output.

Blood Flow Through the Periphery

A. Conductance: Tissues without circulating blood have a low or negligible power of thermal conduction. In fact, experimental work[12] indicates that the body tissue is a good insulator. Virtually all the heat reaching the body surface is brought by the blood and so the heat flux becomes an index of blood flow through the surfaces. The quantity of heat which must be eliminated depends on the metabolism of the tissues, which, of course, is determined by the oxygen consumption. *Conductance of the Superficial Tissues* is a concept that has been used by both Hardy and DuBois[13] and Winslow et al,[14] and has been formulated to include the factors mentioned. The formula of Winslow is as follows:

$$c = \frac{M \pm S}{A\,(T_B - T_S)}$$

where M is heat produced and S is heat stored, both expressed in kilogram calories per unit time. A is the area of the body as designated by the DuBois formula.[15] T_B is body or rectal temperature (Centigrade). T_S is

[12]The Physical Laws of Heat Loss from the Human Body, by J. D. Hardy. (*Proceedings of the National Academy of Sciences*, 23, 631, 1937.)

[13]Loc. Cit. See Note 4.

[14]Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Physiology*, 120:1, 1937.)

[15]A Formula to Estimate Approximate Surface Area If Height and Weight Be Known, by E. DuBois and E. F. DuBois. (*Archives of Internal Medicine*, 17:863, 1916.)

Table 2—Peripheral Blood Flow—Conductance

Subject	Condition	Dry Bulb Temperature, F	Wet Bulb Temperature, F	Oxygen Consumption cc/min	Mean Skin Temperature, F	Rectal Temperature, F	$T_R - T_S$	Conductance Kilogram Calories/F/min	H_2O Loss, Grams/min
		84.5	64.5	.....	93.7	98.4	4.7	.....	0.442
		84.0	64.6	185.0–178.5	92.4	97.5	5.1	0.17	0.438
	Comfortable	83.4	62.0	.....	92.3	98.2	5.9	.....	0.511
		84.0	63.5	188.5–187.5	92.8	.....	.....	.....	0.442
		84.0	61.7	178.3–190.5	92.7	97.9	5.2	0.17	0.460
		Average		184.7	92.8	98.0	5.3	0.17	0.459
J. Morgan (Female)	Hot and Dry	99.4	65.0	185.5–188.3	96.8	98.8	2.0	0.45	2.234
		99.4	67.5	181.3–183.0	96.3	98.4	2.1	0.42	2.235
		99.8	71.9	.....	96.7	98.9	2.2	.....	2.584
		99.5	71.6	185.8–182.2	96.2	98.4	2.2	0.40	2.457
		Average		184.4	96.5	98.6	2.1	0.42	2.378
	Hot and Wet (No fever)	98.7	84.5	176.7	97.3	98.8	1.5	0.57	2.777
		99.5	81.5	.....	96.9	98.8	1.9	.....	2.362
		99.5	81.0	.....	97.1	98.6	1.5	.....	2.236
		Average		176.7	97.1	98.7	1.6	0.57	2.458
	Hot and Wet (Fever)	98.8	86.1	193.3	97.8	99.2	1.4	0.67	2.313
		99.0	87.2	176	97.6	99.2	1.6	0.53	2.841
		99.5	90.0	215.3	98.0	99.5	1.5	0.69	3.774
		Average		194.9	97.8	99.3	1.5	0.63	2.976
	Comfortable	85.8	63.3	268–252	92.9	97.4	4.5	0.28	1.36
		86.0	63.5	272	93.8	98.2	4.4	0.30	1.33
		87.0	65.8	286	93.8	97.9	4.1	0.34	1.28
		Average		269	93.5	97.8	4.3	0.30	1.32
		99.5	64.5	254.0–245.3	96.6	98.5	1.9	0.64	3.30
T. Schertz (Male)	Hot and Dry	99.5	68.1	.....	96.4	98.2	1.8	.....	.....
		99.5	70.0	251	96.9	98.6	1.7	0.71	3.46
		99.0	73.0	.....	96.8	98.5	1.7	.....	.....
		99.5	75.0	.....	96.9	98.5	1.6	.....	.....
		Average		250.1	96.7	98.5	1.7	0.71	3.38
	Hot and Wet	100.0	84.0	285–282	97.3	99.2	1.9	0.72	5.81
		98.7	84.8	.....	97.1	99.3	2.2	.....	5.20
		98.8	85.0	.....	97.3	99.4	2.1	.....	5.50
		99.3	86.0	266.2	97.9	99.4	1.5	0.85	.....
		Average		277.7	97.4	99.3	1.9	0.71	5.47

mean weighted skin surface temperature (Centigrade). In Table 2 the *conductance values* for the two subjects are shown. In computing these values the following formula has been used:

$$C = \frac{M}{T_R - T_S}$$

where M is total metabolism of the subject expressed in kilogram calories per minute, T_R is rectal temperature, and T_S is weighted mean body surface temperature in Fahrenheit.

The heat loss in the conductance formula was taken to be equivalent to the heat production as measured by oxygen consumption. This is, of course, an approximation, which is justified by the absence of change in rectal temperatures. Storage therefore, may be neglected. The conductance is figured for the subject as a whole rather than for a unit area. Hence the factor A is omitted. Conductance values derived by this formula for an individual cannot be compared with values from other individuals.

It is apparent at a glance that the conductance or the peripheral blood flow rises as the mean skin temperature rises. This rise from the comfortable to the hot dry atmosphere is of considerable magnitude, and would indicate that the peripheral blood flow was increased 250 per cent. The increase in conductance from a hot dry to a hot wet atmosphere is not so great (35 per cent) in subject Morgan and even less in subject Schertz.

B. Venous Oxygen Saturation: Table 3 presents venous oxygen saturation results on five healthy male subjects. This measurement has been used in the past to study rate of blood flow through an extremity in man.[16] Since oxygen consumption, or heat produced in an extremity, does not change under different environments, the oxygen saturation figures on venous blood become an index of rate of blood flow. Arterial blood normally is 95 per cent saturated with oxygen. Under the comfortable conditions the venous blood bears about 70 per cent saturation with considerable individual variation. This was observed to be elevated significantly in four of the five subjects under hot dry conditions. Under hot wet conditions the saturation of venous blood approximates arterial blood in three of the five subjects. Apparently the circulation through an extremity can exceed in extreme heat eight times that under comfortable conditions.

These data supplement the data on conductance in that the hands (probably the feet) can open capillary beds widely and tolerate a maximal rate of blood flow through

[16] Intermittent Venous Hyperemia in the Treatment of Peripheral Vascular Disease, by G. DeTakats, F. K. Hick and J. C. Coulter. (*Journal American Medical Association*, 108:1951, 1937.)

Table 3—Peripheral Blood Flow—Venous Oxygen Saturation

Subject	Condition	Dry Bulb, Temperature, F	Wet Bulb, Temperature, F	Venous Oxygen Saturation, Per Cent	Hemoglobin, Grams per 100 cc
G. Brebis	Comfortable	84.8	56.5	54.8	16
	Hot Dry	99.5	62.7	89.9	14.6
	Hot Wet	98.8	90.8	91.0	14.4
Gregoric	Comfortable	84.7	56.5	74.5	13.2
	Hot Dry	99.5	62.7	91.4	13.3
	Hot Wet	98.8	90.8	93.4	14.2
McKeever	Comfortable	84.7	56.5	65.8	13.5
	Hot Dry	99.5	62.7	81.2	14.1
	Hot Wet	98.8	90.8	93.0	14.8
Carlson	Comfortable	82.5	60.0	70.4	15.5
	Hot Dry	99.2	62.2	73.2	16.6
	Hot Wet	99.5	90.3	84.0	16.5
Schertz	Comfortable	81.5	62.0	66.0	13.8
	Hot Dry	99.5	64.5	77.2	.....
	Hot Wet	.....	.....	81.0	.....

Table 4—Blood Volume Studies

Subject	Condition	Dry-Bulb, Temperature, F	Wet-Bulb, Temperature, F	Plasma Volume, cc	Circulating Blood Volume cc	Hemato-crit	Pulse
J. Morgan	Comfortable	83.4	62.0	2342	3747	37.5	63
	Hot and Dry	99.8	71.9	2453	3882	36.8	70
	Hot and Wet	98.7	84.5	2778	4375	35.5	74
T. Schertz	Comfortable	81.5	62.0	4651	8189	43.2	48
	Hot and Dry	99.5	68.1	4808	8108	40.7	50
	Hot and Dry	101.0	75.5	4902	8239	40.5	49
	Hot and Wet	100.0	83.0	5263	8831	40.4	56

them. The conductance values mentioned do not show such a great change in blood flow, because they are derived from a mean skin temperature weighted to include the surfaces of the body which cannot tolerate excessive flow of blood through the skin. In general the skin over the trunk, thighs, and shoulders does not have the number of available capillaries to open during hot exposure which are found in the skin of the hands.

Water loss is shown in Table 2 in grams per minute. In the hot dry condition where the sweat produced approaches the amount lost in the hot wet condition, the skin of the female subject was usually dry to the touch and definitely cool. The male subject produced excessive quantities of sweat in the hot wet environment, much of which ran off without evaporating. He, too, usually had most of his body dry to the touch while in the hot dry condition. Obviously the quantity of water lost through sweating bears no constant relationship to the rate of blood flow through the skin as measured by conductance or venous oxygen saturation. There is some relationship of heat loss by evaporation and circulation through the skin until free sweating appears.

C. Blood Volume: Blood volume studies have been completed on a male and female subject under hot, wet and hot, dry, as well as comfortable conditions. The results are presented in Table 4. There is an observed rise of about 8 to 12 per cent in these two subjects in hot, wet conditions. Bazett[17] has observed a rise in circulating blood volume in hot conditions after days of exposure. The results presented in this paper are after only a few hours, two or three, of hot wet conditions. Apparently the body calls on its store of water to raise the blood volume as the capillary bed in the skin dilates. This was independent of fever for the rectal temperature was 99 F in Schertz and 98.8 F in Morgan on the days of the hot, wet conditions.

Discussion of Results

It appears that the rate of blood flow through the skin governs its temperature and that the skin temperature is the delicate mechanism for controlling loss of heat to meet the needs of the individual. The skin being dry or wet, however, depends on the humidity in the presence of a high dry-bulb temperature. With a low humidity in the evaporative zone the skin temperature is lower than with a high humidity, the dry-bulb remaining the same. This has been long well known and accounts for the ability of men to work in deserts, and the difficulty of hard physical work in the tropics.[18]

That the cardiac output does not rise as soon as the pulse rate is a matter of considerable interest. It would appear that the body *should* possess a mechanism of increasing the cardiac output to facilitate greater movement of blood through the skin in the presence of great heat. The data do not show this, however. These observations are quite limited in number but appear significant.

Christensen[19] found that in work cardiac output is a function of the heat production and does not rise until heat production rises. Data resulting from the studies reported here support the concept that with rising environmental temperature the skin temperature rises as a result of deflection of blood through the skin. This does not entail an increased cardiac output and may not require an increased pulse rate. In fact the pulse rate may rise without the cardiac output rising, which constitutes a type of inefficient heart action.

As the rectal temperature rises the heat production rises and cardiac output goes up.

The requirement of heat loss does not dictate cardiac output. It does dictate mean skin temperature and rate of blood flow through the skin.

Conclusions

The cardiac output does not rise on exposure of a resting man to hot conditions until his metabolism (heat production) rises.

The blood flow through the periphery, particularly the hands, becomes very great as the mean skin temperature approaches the rectal temperature.

The blood volume rises within periods of two or three hours exposure to hot, wet conditions.

Cardiac output is governed by heat production.

The skin temperature can be duplicated from one day to another in controlled rooms. It is the delicate mechanism governing heat loss. In the evaporative zone it is partly governed by humidity.

Venous oxygen saturation in superficial veins is an index of rate of blood flow through an extremity. Figures as high as 93 per cent of saturation were observed in hot wet conditions.

Acknowledgment

Acknowledgment is made of the cooperation of the Engineering Experiment Station of the University of Illinois, Urbana, Ill., through engineering consultation and services rendered by Prof. A. E. Hershey and Prof. M. K. Fahnestock of the Department of Mechanical Engineering.

[17] Cardiovascular Changes in a Hot Room, by H. C. Bazett, J. C. Scott and F. W. Sunderman. (*American Journal of Physiology*, 123:11, 1938.)

[18] Life, Heat and Altitude, by D. B. Dill. (Harvard University Press, Cambridge, 1938.)

[19] Arbeitsphysiologie, by E. H. Christensen. (5:479, 1932.)

New Headquarters for British Institution

The Institution of Heating and Ventilating Engineers of England have announced a change of address to 22 Russell Square, London, W.C.1, England.

Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice

Report of the ASHVE Research Technical Advisory Committee on Physiological Reactions: C.-E. A. Winslow, *Chairman;* T. Bedford, E. F. DuBois, R. W.Keeton, A. Missenard, R. R. Sayers, and C. Tasker

Introduction

SUBSTANTIAL advances have been made in recent years in the understanding of the fundamental physiological problems underlying the art of air conditioning. It is both unnecessary and impracticable to review here all the researches which have been published in England, France, and Canada, and at the ASHVE Research Laboratory, the U. S. Bureau of Mines, Cornell University, Harvard University, the University of Illinois, the Russell Sage Institute of Pathology, the John B. Pierce Laboratory of Hygiene, and elsewhere in this country, but it does seem worth while to call attention to a few of the salient points which have been brought out and to indicate certain of the immediate implications of these researches for air conditioning practice.

It may safely be said that the course of physiological and clinical studies has consistently tended to emphasize the problem of heat interchange between the body and its environment as of basic and fundamental importance. The search for mysterious chemical and electrical relationships has continued, but to date there is no valid evidence that any such influences are of material significance under ordinary living conditions. All carefully conducted work on the physiological effect of ions has been negative and there is nothing to encourage the wider usage of ionization.

Ozone has been used with success for the destruction of micro-organisms (molds) in meat packing establishments and the like; and where considerable amounts of organic effluvia are present this substance may be useful as a deodorant. For ordinary ventilation practice, however, neither of these purposes can be usefully attained, since the concentration of ozone necessary for effectiveness would be likely to transcend the limit of comfort in ordinary occupied rooms. While ozone has been used in the treatment of certain diseases, there is no evidence that it has a tendency to increase comfort or to benefit health under conditions of normal human occupancy. Little or nothing has been added to the old conclusions of the New York State Commission on Ventilation that in the school room the use of ozone would be objectionable rather than desirable. The practice had its advocates for a period in St. Louis, but the expensive apparatus and installation has been completely abandoned.

It should be noted that this report deals with physiological reactions to the air of ordinarily occupied rooms and does not consider the removal of industrial dusts or the destruction of germs of communicable diseases in the atmosphere. These problems fall within the scope of other committees.

For presentation at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

Zones of Physiological Response

So far as physiological reactions in indoor spaces, under conditions of normal human occupancy, are concerned, the basic problems are thermal in nature. Perhaps the first point of importance brought out by recent studies is the fact that the reactions involved in a cold and in a hot environment are radically different in nature. Therefore, any attempt to formulate simple engineering relationships covering the entire thermal scale are obviously doomed to failure. The most complete information available relates to the lightly clothed subject[1] in a semi-reclining position (John B. Pierce Laboratory). These experiments indicate that the clothed, semi-reclining subject exhibits normal and easy physiological regulation between air temperatures of 77 and 86 F (25 and 30 C). Above 86 F (zone of evaporative regulation) the individual adapts by active sweat secretion. Below 77 F the skin temperature falls, but adaptation is incomplete and the temperature of the body tissues falls more or less continuously (zone of body cooling).

Fig. 1 shows physiological changes in these zones for such clothed, semi-reclining subjects. In the zone of evaporative regulation, metabolism remains constant and increasing evaporative heat loss just balances decreasing heat loss and increasing heat gain (above the zero line), due to rising operative temperature. In the zone of body cooling, metabolism rises a trifle, below 68 F (20 C), evaporative heat loss falls very slightly, and storage (cooling of the body tissues) rises sharply as heat loss due to convection and radiation increases.

The conception of *storage* is essential for an understanding of the reactions of the human body to its environment. As used in this report, it implies a rate of cooling (or heating) of body tissues and, according to convention, positive storage involves cooling, and negative storage, heating, of the tissues. This process, if cooling is rapid, cannot be measured by rectal temperature, since a change in the temperature of the circulating blood lags far behind the cooling of the skin and the muscle masses in the limbs and other organs. Furthermore, the process may continue for a long period; in certain recent experiments, both skin temperature

[1] A two-piece suit of cotton underwear, a cotton shirt without tie, socks, low leather shoes, and a dark gray, single-ply suit with three-quarter lined coat and fully lined vest.

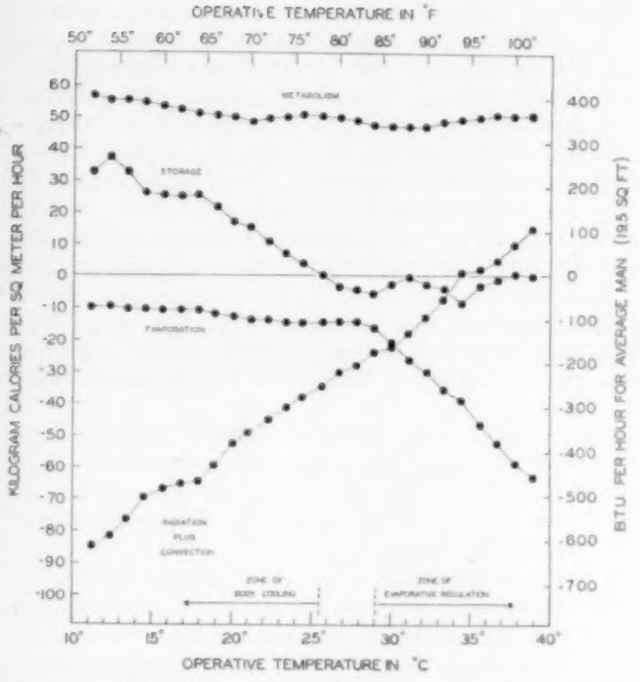
OPERATIVE TEMPERATURE IN °F
50° 55° 60° 65° 70° 75° 80° 85° 90° 95° 100°
METABOLISM
STORAGE
EVAPORATION
RADIATION PLUS CONVECTION
ZONE OF BODY COOLING
ZONE OF EVAPORATIVE REGULATION
KILOGRAM CALORIES PER SQ METER PER HOUR
BTU PER HOUR FOR AVERAGE MAN (19.5 SQ FT)
OPERATIVE TEMPERATURE IN °C
10° 15° 20° 25° 30° 35° 40°

Fig. 1—Relation between metabolism, storage, evaporation, radiation plus convection, and operative temperature for the clothed subject

and rectal temperature were found to be still falling after a period of six to eight hours on exposure to a cold environment.

Reactions in the Cold Zone

An interesting point indicated by the studies at the John B. Pierce Laboratory is that when a given operative temperature (that is, the temperature which should physically exert a certain calorie-demand upon a body of fixed surface temperature) is produced by (*a*) air and walls at approximately the same temperature, (*b*) colder air and warmer walls (in the zone of body cooling), the skin temperature falls to a lower point for the cold-air warm-wall situation, thus decreasing the actual rate of heat loss. The reason for this effect is somewhat obscure, but it is believed to be related to local stimulation by the colder air exerted on the membranes of the nose and throat and to the greater chilling of the exposed skin surfaces when those parts of the body are moved. Studies made at the John B. Pierce Laboratory in this connection indicate surprising influences on decrease in skin temperature exerted by brief cold stimulation of small regions of the body, as by an ice bag on the neck. This is of some practical interest as indicating that warm walls with cool air tend to promote vaso-motor adjustment of the body to the thermal environment, and that extreme local chilling, as by localized drafts, may well be harmful.

An important point which has been brought out in these studies is the very minor effect of variations in relative humidity in the zone of body cooling. Below an air temperature of 77 F (25 C)—for the clothed, semi-reclining body—sweat secretion is minimal (only a quarter of the total heat loss) and variations downward in the relative humidity of the atmosphere exert only a minor effect upon evaporative heat loss. The reason for this is that the vapor-pressure at the skin surface (at 91 F, or 33 C) is 1.47 in. of mercury; while in air at 70 F (21 C), 80 per cent saturated, it is only 0.58—a gradient which permits full evaporative heat loss. Under conditions of low air movement, and with a relative humidity of 30-35 per cent, the evaporative heat loss is about 4 kilogram-calories per square meter (1.5 Btu per square foot) per hour greater than with a relative humidity of 75-80 per cent. This is the equivalent of increasing the air temperature by less than 1 F.

Excessive atmospheric humidity at low temperatures may exert a chilling effect through increased conductivity of the clothing. General empirical impressions indicate that this factor may be an important one, but it has not as yet been sufficiently studied under laboratory conditions.

Heat losses from the body in the cold zone are, then, influenced chiefly by air and wall temperatures, and very markedly by air movement. The effect of the quantitative influence of air movement has not as yet been fully worked out, and long-time experiments with this end in view are now in progress. It is known, however, that air movement greatly increases the heat loss in a cold environment. Recent studies at the John B. Pierce Laboratory indicate that at an air temperature of 70-72 F (21-22.5 C) an increase in air movement from 17 to 100 linear feet per minute is approximately equivalent to a drop of 7 deg in operative temperature. It is generally assumed, though without exact physiological observations, that moderate and variable air movement is favorable to comfort. An air temperature of 72 F (22.5 C) is definitely comfortable with an air movement of 17 fpm; definitely uncomfortable with movement of 100 fpm. It has generally been found that 50 fpm is an upper limit of comfort for persons at rest.

Results for *storage*, as here cited, refer to the decreasing temperature of body tissues over a two-hour period. It is obvious that this process cannot continue indefinitely, and members of the Committee are now conducting long-time experiments to determine how and when a state of thermal balance between the body and its environment may ultimately be reached.

Reactions in the Hot Zone

Passing to the hot zone (above 86 F-30 C for the clothed, semi-reclining body) an entirely different relationship exists. Until the upper limit of adjustment is reached, sweat secretion—through a very perfect physiological adaptation—increases steadily to produce the evaporative heat loss necessary to maintain heat balance between the body and its environment. This process, although successful as a regulatory factor, may be accompanied by symptoms of marked discomfort.

It is important to note that this automatic regulation of thermal interchange in the hot zone is accomplished with remarkable success, however the theoretical heat loss demand may be modified by varying conditions of relative humidity and air movement as well as of operative temperature. It has, however, a very definite upper limit at the point at which the sweat secretion accumulates and runs off the body without exerting further cooling effect. Studies at the John B. Pierce Laboratory have established upper limits for the clothed, semi-reclin-

Table 1—Upper Limits of Evaporative Regulation

Environmental Temperature		Relative Humidity Per Cent
F	C	
126.9	52.7	0
122.0	50.0	5
113.0	45.0	18
108.5	42.5	26
104.5	40.0	38
99.5	37.5	51
95.0	35.0	69
90.5	32.5	89
87.8	31.0	100

ing body at an air movement of 17 linear feet per minute as given in Table 1.

These figures should be of interest for comparison with the very valuable studies being made by another Society Research Committee on Air Conditioning in Industry with regard to the upper limits of temperature and humidity which are bearable in the course of various kinds of industrial work. It should be noted that the data cited apply to a resting position with a low metabolic rate, and the quantitative limits of the various zones must certainly vary directly with changes in metabolism.

It should be understood that such figures as those cited may not necessarily represent the limit of human endurance. When a point is reached at which evaporation of sweat can no longer dissipate the heat produced by metabolism, adjustment may be reached (particularly in the acclimatized subject) by an increase in skin temperature and by negative storage. A. Missenard, in observations on nude workers in a potash mine in France, has found that they were able to perform light work for six consecutive hours with an air temperature of 110.5 F (43.6 C) and a wall temperature of 114.5 F (45.8 C) and 40 per cent relative humidity. At high air temperatures, the effect of an air movement of 100 fpm is to decrease discomfort (from its effect on evaporation), even when the air temperature is actually above the temperature of the skin.

In practice, one of the most important problems of air conditioning is stratification of temperature. It is probably important for comfort as well as for heat economy to make the differential between floor and ceiling as small as possible.

Just as the effect of cold air may be balanced by that from hot walls, so Dr. C. A. Mills has pointed out that hot air may be made comfortable by the counter-effect of radiation to cold walls. The quantitative details have not as yet been worked out but are at present under study by committee members. It should be pointed out, however, that there is no reason to expect any particular physiological values to be introduced by a situation of this kind, while a combination of hot air and artificially cooled walls would be one which would be highly costly to maintain in practice.

Calorimetric Experiments on Nude Subjects

The results reported, from experiments with clothed subjects at the John B. Pierce Laboratory of Hygiene at New Haven, are in substantial accord with a series of researches conducted at the Russell Sage Institute of Pathology in New York. These observations have been carefully made in a calorimeter which records simultaneously the heat production, the heat losses, the vaporization, radiation, and convection. The subjects' notes of mental state are also preserved. Studies have been made on effects of fall in body temperature and on normal chill mechanism, and with alcohol, the effects of hot and cold drinks, and the sensory thresholds to hot and cold stimuli.

In all work on air conditioning it is of prime importance to make proper allowance for the clothing and activity of the subjects. This is brought out in Fig. 2. The first three diagrams are based on studies in the respiration calorimeter showing the physical and physiological changes with rising temperature. The fourth is based on only two experiments. The fifth and sixth are largely extrapolations, as the apparatus was not suitable for measurement of metabolism more than three times the basal level.

It is suggested that diagrams of this type may serve as a basis for a study of the physiology of air conditioning. For example, an analysis of the diagrams shows the following.

I. Reactions of Body to Different Environmental Temperatures. For Nude Subjects Lying in Bed.
   1. Cold Zone 83 F (28 C) and below.
      No control of heat loss; body temperature maintained by chills.
      Blood vessels of skin are maximally constricted throughout the zone; body cools as a physical object in accord with Newton's Law.
      Evaporative loss = one-half respiratory and one-half seepage through skin.
      = constant.
      Chills brought on periodically when mean body temperature has fallen 1.2 F (0.65 C). Alcohol anesthetizes chill mechanism. Chill with alcohol occurs after a degree or more fall in temperature.
   2. Zone of vasomotor control 83 F-68 F (28 C-31 C) (Comfort Zone).
      Subjects most comfortable; no progressive changes in body temperature. Blood flow to the skin changes so that heat loss is automatically adjusted to the environment. No generalized feelings of warmth or cold; no sweating. Chills are easily evoked with cold drinks. Sweating is evoked by hot drinks.
   3. Zone of vasomotor and evaporative regulation 88 F-98 F (31 C-36 C).
      Peripheral blood flow increases in proportion to temperature. Sweating (at constant low humidity), in proportion to temperature. Body cooling is adequate; general level of body temperature is higher, pulse rate increases in proportion to temperature. Body is continually stimulated by sensations of heat which are uncomfortable, but which are necessary to thermal regulation.
      At higher temperatures, even though the body may be able to cool itself from a physical standpoint, reactions to heat may be anomalous, causing heat stroke and exhaustion.

II. It is important to note that a rise in skin temperature of 0.0025 C will evoke a sensation of warmth in three seconds, and a fall in skin temperature of 0.004 C in one second will produce a cold sensation. The peripheral sense of heat or cold is not changed by exposing more of the body if the head is once exposed. The head (face and neck) are the most sensitive portions of the body to thermal stimuli. The backs of the hands are next. Calorie for calorie, *cold* radiation causes twice the skin temperature change that is produced by *heat* radiation. Local vaso-

motor response to thermal stimuli occurs after two minutes at points stimulated. The result is that cold radiation causes a three- to four-fold greater change in exposed skin temperature than does heat radiation.

The threshold for pain is lower for cold than for heat and of a different quality. Various types of heat radiation have different sensory thresholds.

The following differences are noted between men and women. In the cold zone the thickness of the thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone the threshold to sweating is higher and skin temperature levels are also higher.

Problems of Comfort

It will be noted that the operative temperatures cited for thermal equilibrium appear distinctly high as compared with the commonly accepted standard of 68-70 F (20-21 C) for schools and similar occupied spaces. It should be expected that in the long run the comfort zone must coincide with the area of thermal adjustment with minimum physiological effort, and this adjustment, as noted, is reached for the clothed subject in a semi-reclining position, at approximately 80 F (26.7 C) operative temperature, corresponding to 74 F (23.3 C) effective temperature (with 50 per cent relative humidity).

In the very valuable field studies made by the Society at Pittsburgh, Minneapolis, Toronto, and Texas A. & M. College in regard to summer cooling requirements, effective temperatures of 71 F (21.7 C) or thereabouts, corresponding (with 50 per cent relative humidity) to an operative temperature of 76 F (24.4 C), seem to be preferred.

On analysis these results do not seem inconsistent with each other, or with earlier practice. Both physiological adjustment and sensations of comfort will vary with three factors; the heat production of the organism (metabolism), the clothing worn, and the acclimatization associated with the season of the year.[2] Such results as those obtained at the Pierce Laboratory, which indicate an optimum around 80 F (26.7 C) operative temperature, will obtain in a semi-reclining position, with metabolism at a low level and with light clothing. The fact that the

[2]Reference is here, of course, made not to basal metabolism (minimal) which is relatively constant, but to the actual mean daily metabolism as affected by the activity of the subject.

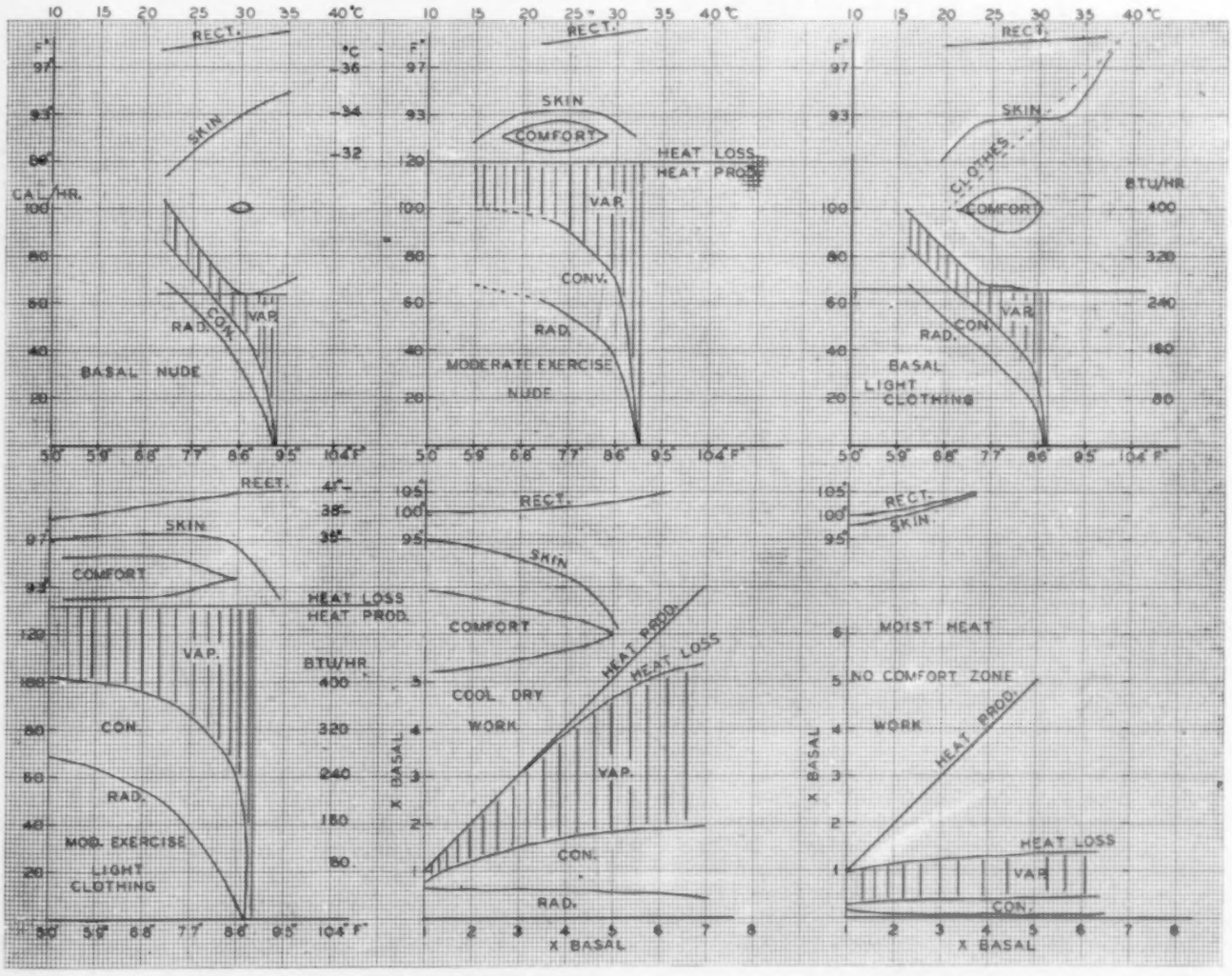

Fig. 2—The relation of atmospheric temperature and metabolism to various types of heat loss and comfort for the nude subject

In the first four diagrams, the ordinates represent rectal and skin temperatures (above) and calories of heat production and heat loss (below); the abscissae represent air and wall temperatures, which were identical. Heat loss is divided into radiation, convection and vaporization. In the last two diagrams, the ordinates represent heat production and heat loss using the basal (minimal) metabolism as a unit equal to one; the abscissae represent activity of the subject in terms of basal metabolism. Twice the basal value corresponds to walking slowly and six times the basal value to heavy labor.

ASHVE experiments on summer cooling were made with subjects having a substantially higher metabolism (since they were at work and not reclining), should account for the 4 F (2 C) lower optimum observed. Earlier observations in school rooms pointing to a desirable operative temperature of 70 F (21.1 C) or below were obtained with a physical activity more or less comparable with that of less active factories, with relatively high metabolism on account of the fact that the subjects were children, and with heavy clothing, and acclimatization to winter conditions. (All these data are associated with low air movement.)

It is interesting to note that the New York State Commission on Ventilation found operative temperatures below 55 F (182. C) preferred by students in a gymnasium. It seems of fundamental importance that the comfort zone may vary over a range of 25 F (14 C), depending on the physical activity, the clothing and the acclimatization of the subject.

Methods of Air Examination

In connection with studies of atmospheric conditions from the viewpoint of human health and comfort, attention should be called to the following recommendations of the Sub-Commission on the Hygiene of Environmental Conditions in the Dwelling, of the Housing Commission of the League of Nations, adopted at Geneva on June 25, 1937:

"Valuable as these various instruments[3] may be in establishing the conditions of comfort in any given enclosed space, it is of great importance in all research studies to make an accurate record of each of the four independent factors governing bodily heat exchange, temperature, movement and humidity of the air, and mean radiant temperature of the surrounding surfaces. For this purpose we suggest in the interest of comparability the use of the following four types of instruments or others yielding similar data:

(*a*) Silvered dry-bulb thermometers or hair-pin thermometers (Bargeboer).

(*b*) Silvered dry Kata thermometers or the hot wire anemometer.

(*c*) Psychrometer, wet- and dry-bulb, whirling or ventilated.

(*d*) Globe thermometer (Vernon) or the dry resultant thermometer (Missenard).

Such instruments as these, when properly calibrated and their readings are compared, can be used for determining the four basic physical factors concerned separately or in certain combinations. The results of the four physical measurements thus determined can generally be translated into the terms of any special instrument combining two or more of them.

The single indices of thermal conditions mentioned above may be found to be of great utility for discussions in the countries of their adoption. It would, however, facilitate comparisons with the scales of measurement used in other countries if the measurements of the separate thermal factors were published along with those of the single index.

In some instances it may be important to record not only the movement and temperature of the air at various levels, but also the temperature of each wall and window, of the flooring, and of the ceiling, and to measure the total effective radiation of the surroundings in six directions—in order to trace the exact causes of defects in the building which have an unfavorable influence on the heat exchanges of its inhabitants. Facts of this type are of great practical importance."

The ASHVE Technical Advisory Committee on Physiological Reactions endorses these viewpoints and urges that in all studies of air conditions records be obtained showing:

(*a*) True air temperature (free from radiation effects).

(*b*) Air movement.

(*c*) Humidity.

(*d*) Mean radiant temperature of surrounding surfaces.

[3]Refers to various instruments (such as the eupatheoscope and the thermo-integrator) designed to measure the combined influence of two or more of the factors of air temperature, wall temperature, relative humidity, and air movement.

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Some Reflection and Radiation Characteristics of Aluminum

By Cyril S. Taylor* and Junius D. Edwards** (*MEMBER*), New Kensington, Pa.

ALUMINUM has proved to be an excellent reflecting material for radiation in all parts of the spectrum. Its high reflectivity for both visible and ultra-violet radiation, combined with its non-tarnishing characteristics, have won its adoption for coating reflectors for telescopes and other optical instruments. The development of the aluminum reflector,[1] which is given both brightness and permanence by a duplex electrolytic treatment, has afforded aluminum wide application in the lighting field. The high reflectivity and correspondingly low emissivity of aluminum for infra-red radiation have been responsible for the application of aluminum foil in insulating structures. While considerable data have been accumulated as to the performance of aluminum in the visible range, there is less known about its behavior with infra-red radiation, and particularly the effect of surface films upon its reflectivity and emissivity.

The measurement of radiation from surfaces at relatively low temperatures (approximately 100 F) presents a number of difficulties. Several satisfactory emissivity meters have been described in technical literature,[2,3] but apparatus for such measurements is not generally available from instrument makers. In this paper there is given a description of an instrument suitable for making measurements of the emissivity of surfaces at temperatures up to 300 F or higher, together with a variety of measurements on aluminum obtained by the use of this instrument. The general plan of this instrument follows closely that of an emissivity meter designed by R. E. Gould, to whom acknowledgement is made. Advantage was taken of a number of details of design described by other workers, and no special originality is claimed for the equipment.

Description of Emissivity Meter

The emissivity meter comprises a heater which heats the sample being measured to a definite uniform temperature, a radiometer which receives energy radiated from the specimen, and a potentiometer system by which the electromotive forces generated by the radiometer and various essential thermocouples are quantitatively measured. The general construction and operation of the meter can best be described by reference to Fig. 1, which shows a cross-section of the instrument.

The specimen holder and heater unit (*E*) is placed on top facing downwards, so as to eliminate convection currents as far as possible. The heater face-plate is a piece of aluminum sheet, oxide-coated. It is heated by another sheet, oxide-coated for insulation, wound with nichrome ribbon and backed by a similar-sized insulating block, pressed against it and bolted together. A copper-constantan thermocouple, embedded in a small groove cut in the face of the heater in such a way that the junction (*J*) is located at the center of the face, permits the temperature of the hot face to be measured. The heater, which

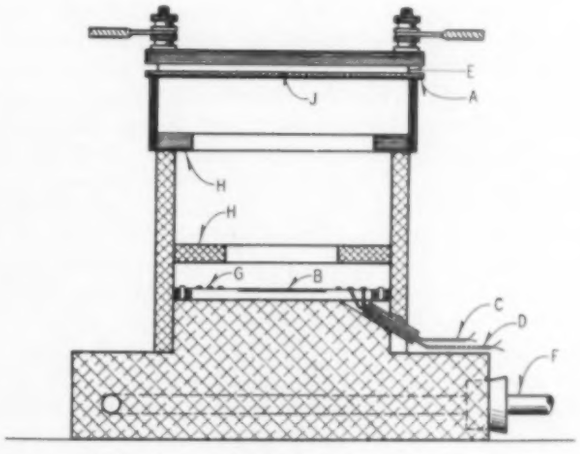

Fig. 1—Sectional view of emissivity meter:

A. Sample for emissivity measurement
B. Thermopile receiving radiation from *A*
C. Thermocouple leads measuring temperature of thermopile cold junctions
D. Thermopile leads
E. Electric heater plate for maintaining sample at uniform temperature
F. Inlet for constant temperature water supply circulating in base
G. Copper pins to which are attached cold junctions of thermopile
H. Radiation shields
J. Thermocouple junction for measuring sample temperature

is usually operated at a temperature of 100 F, requires about 4w (watts) power. The sample to be measured (*A*) is fastened to the face of the heater by means of a thin coat of cement.

The radiometer consists of a copper-constantan thermopile (*B*) unit of 18 junctions mounted on the central portion of a water-cooled (*F*) aluminum block. The central cylindrical part of the block is $3\frac{1}{16}$ in. in diameter and projects $\frac{3}{4}$ in. above the rest of the block, which acts as a base. A tight-fitting, thick-walled aluminum tube is forced onto the projecting cylinder. This tube extends upwards, 2 in. above the thermopile, and has fastened in it, $\frac{3}{8}$ in. above the thermopile, a radiation shield (*H*) consisting of an aluminum ring $\frac{1}{4}$ in. thick and with a circular opening $1\frac{5}{8}$ in. in diameter. On top of the aluminum tube is fastened another radiation shield (*H*)

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[1]Aluminum for Reflectors, by J. D. Edwards. (*Transactions, Illuminating Engineering Society*, 29, p. 351, 1934.)
[2]An Apparatus for the Measurement of the Total Normal Thermal Emissivity of Sheet Materials in the Temperature Range 140 to 500 F, by Paul F. McDermott. (*Rev. of Sci. Instruments*, 8, 1937, p. 185.)
[3]Measurement of Radiation from Aluminum Surfaces, by E. Schmidt. (*Hausseit. V.A.W. Erftwerk A.G. Aluminium*, 2, 1930, p. 91.)
For presentation at the 45th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1939.

made of a synthetic resin ring, ¼ in. thick and with an opening having a diameter of 2½ in. The exposed bottom of this ring is coated with aluminum foil to reduce as much as possible any radiation from this source. A thin tube of synthetic resin, approximately the diameter of the aluminum tube, is mounted on top of the ring to act as a low-heat-conducting support for the heater.

The receiver of the thermopile is a lacquered disk of aluminum foil, 1¼ in. in diameter and 0.0003 in. in thickness, to which are cemented 18 thin copper segments, each of which has soldered to it a thermocouple of No. 40 B. & S. gage copper-constantan wire. The copper segments were cut from a copper disk 0.0003 in. thick and they covered practically the whole area of the receiver. The assembled receiver disk is blackened with smoke from burning camphor. The thermopile leads are shown at (*D*). The cold junctions are soldered to heavy copper pins (*G*) inserted in a synthetic resin ring. A copper-constantan thermocouple, soldered to one of the cold junction pins, allows the temperature of the cold junction to be obtained (*C*). The whole thermopile unit was built and assembled, following very closely the directions given by Heilman.[4]

In calibrating the emissivity meter, a piece of bright aluminum foil was fastened to the heater plate to establish the radiometer reading with a surface of low emissivity. A sheet of black paper was employed to establish the radiometer reading for a point of high emissivity. The aluminum foil was assumed to have an emissivity of 3 per cent at 100 F, and a value of 95 per cent was assigned to the black paper. The radiometer readings for surfaces of intermediate emissivity were established by exposing various proportions of foil and black paper as the radiating surface. This was accomplished by cementing to the aluminum foil pie-shaped sectors of black paper.[5] These were always attached as diametrically opposite pairs of equal size, so as to avoid any possible error resulting from lack of exact centering. In this way a surface could readily be obtained whose average emissivity was any value between 3 and 95 per cent.

In this work the standard of reference for low emissivity was bright aluminum foil with an assigned value of 3 per cent at 100 F. In this paper the emissivity values are expressed as a percentage of the emissivity of a *black body* at the same temperature. There is very good evidence in the measurements of McDermott,[6] H. Schmidt and Furthmann,[7] Coblentz[8] and E. Schmidt[9] that the emissivity is as low as 3 per cent and possibly lower. If another value is eventually established for the emissivity of aluminum foil, the emissivity values reported in this paper can be corrected accordingly. The commercial aluminum foil with an emissivity of 3 per cent has a purity of about 99.4 per cent. Schmidt's data at various temperatures showed an increase in emissivity of one sample of aluminum from 4.0 per cent at 200 C to about 5.0 per cent at 500 C. Recent measurements from this laboratory gave a value of 2.6 per cent for the emissivity of aluminum, 99.996 per cent pure.[9a]

[4]Surface Heat Transmission, by R. H. Heilman. (*Mechanical Engineering*, *51*, 1929, p. 355.)

[5]A Reflectometer for All Types of Reflecting Surfaces, by Frank Benford. (*Gen. Elec. Rev.*, *37*, 1934, p. 457).

[6]Loc. Cit. See Note 2.

[7]Radiation from Solid Bodies, by H. Schmidt and Furthmann. (Mitt Kaiser-Wilhelm Inst: Eisenforsch. Abhandl., *10*, 1928, p. 225.)

[8]Radiometric Investigations of Infra-red Absorption and Reflection Spectra, by W. W. Coblentz. (*Bureau of Standards Bulletin 2*, 1906, p. 457.)

[9]Loc. Cit. See Note 3.

Effect of Oxide Films Upon Emissivity

In general, non-metallic substances exhibit high emissivity properties, and aluminum oxide is no exception to this general rule. It might be expected, therefore, that the oxidation of aluminum surfaces would increase their emissivities. With normal atmospheric oxidation, however, the films are usually so thin as to have little effect upon emissivity. The oxide film, moreover, is always present and all emissivity measurements on aluminum are for a surface covered with an oxide film, even though it be extremely thin. Various estimates of the thickness of the oxide film naturally present upon aluminum are of the order of 1 to 2 x 10^{-6} cm (approximately 0.4 to 0.8 x 10^{-6} in.).[10,11]

The oxide film upon aluminum can be increased in thickness and changed in character by various chemical and electrochemical treatments. The thin dielectric films produced upon aluminum by anodic oxidation, for use in electrolytic condensers, are frequently of appropriate thickness to show interference colors. Substantially thicker films are applied for decorative and protective purposes, and these films in general range from about 0.0001 in. to 0.0006 in. (0.0025 mm to 0.015 mm) in thickness. In Fig. 2 is shown the relationship between emissivity and thickness of film for a series of oxide films electrolytically produced in a 15 per cent sulfuric acid electrolyte at 70 F.

It will be noted from the curve that the emissivity increases linearly with increase in thickness of the oxide coating up to about 8x10^{-5} in. Above about 10x10^{-5} in., the emissivity shows only a slow change with increase in thickness; the emissivity has already reached a high value at this point. All oxide coatings do not show the same emissivity values at the same thickness; the emissivity may be greater or smaller than the values of

[9a]The Properties of High Purity Aluminum, by C. S. Taylor, L. A. Willey, D. W. Smith and J. D. Edwards. (*Metals and Alloys*, *9*, 1938, p. 189.)

[10]Some Optical Observations on the Protective Films on Aluminum in Nitric, Chromic and Sulphuric Acids, by L. Tronstad and T. Höverstad. (*Transactions, Faraday Society*, *30*, 1934, p. 362.)

[11]Second Experimental Report, Section V, Aluminum, by W. H. J. Vernon. (*Transactions, Faraday Society*, *23*, 1927, p. 152.)

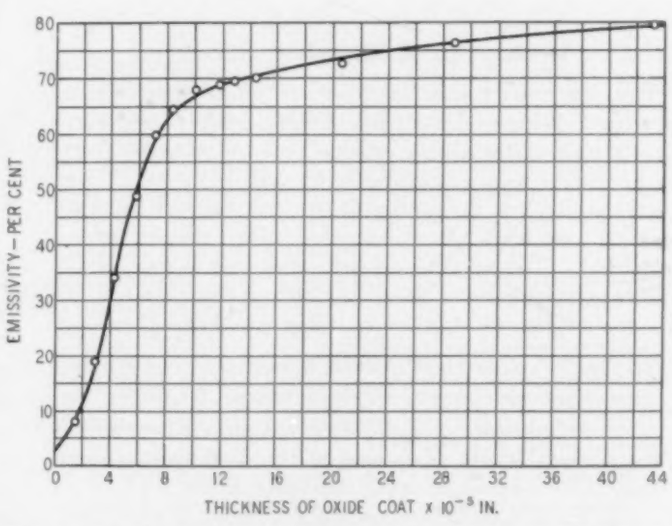


Fig. 2—Relation between thickness and emissivity (at 100 F) of an electrolytically produced oxide coat on aluminum

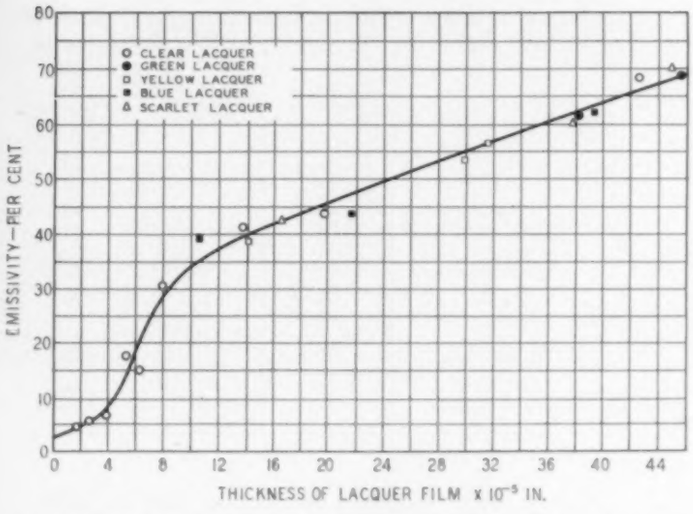


Fig. 3—Relation between thickness and emissivity (at 100 F) of lacquer films, clear and colored with dyes, on aluminum

Fig. 2, depending upon the structural characteristics of the coating, which are controlled by the electrolyte employed and the coating conditions. These differences may be substantial.

The thickness of the oxide coatings shown in Fig. 2 was measured in several ways. When the coating had a thickness of about 10 x 10^{-5} in. or greater, it was easy to measure the thickness directly by observing a cross-section of the foil and coating under the microscope. The technique of preparing the sample for examination is described in an earlier report.[12] Some very thin oxide coatings showed interference colors. In these cases it was possible to measure their thickness by the method of Blodgett and Langmuir,[13] using as color standards a series of built-up films of known layers of barium stearate on polished chromium. Dr. K. B. Blodgett kindly made such a comparison and estimated the thickness of the oxide films on two samples of aluminum foil, which had been electrolytically oxidized in boric acid electrolyte, to be 19 and 26 x 10^{-6} cm, respectively. The thickness was also evaluated by measuring the increase in weight resulting from oxidation and calculating the thickness from the measured area and the density of the oxide. This method gave thickness values of 19 and 27 x 10^{-6} cm, which are quite comparable with those measured by Dr. Blodgett on similar samples.

According to the Wien displacement law, the energy peak for surfaces radiating at temperatures of about 100 F occurs at a wave-length of about 9.3μ (0.009 mm). Apparently, therefore, the oxide-coated surface only behaves like the massive oxide when the coating thickness reaches a value approaching the same magnitude as that of the wave-length of the radiation being reflected or emitted.

The Emissivity of Lacquer-coated Aluminum

In Fig. 3 is given a series of data showing the emissivity of aluminum foil coated with lacquer films of various thicknesses and colors. The lacquer in all of these films was formulated with a certain pyroxylin base to which had been added a small amount of resin. For the colored films, various dyes were added to give scarlet, yellow, green and blue films. The presence of the dye seemed to have little effect upon the emissivity as compared with the clear coating, and there was no specific effect due to any of the particular colors. By comparing the emissivity values of Fig. 3 with Fig. 2, it will be observed that the lacquer coatings in general have less effect in increasing the emissivity than does an oxide coating of similar thickness.

Other lacquer films were observed in which the color was obtained by incorporation of a pigment. In general, the aluminum foil bearing these films showed emissivities intermediate between the foils with the dyed lacquer films and the oxide films. The thicknesses of the lacquer films were measured by determining the increase in weight of the coated foil, caused by the presence of the lacquer, and calculating the thickness from the density of the dried lacquer.

It is quite practical to coat foil commercially with a uniform, thin film of lacquer which does not raise the emissivity above a value of 10 per cent. Such a coating of proper composition offers substantial protection against deterioration where the service is such that prolonged contact with moisture is anticipated.

Reflectivity for Visible Light

While the eye may be a good judge of the reflectivity of a surface for light, it cannot always appraise the value of a surface for reflecting infra-red radiation. This can be demonstrated by a few selected examples. In the case of the data on oxide films shown in Fig. 2, the emissivity increases from a value of 3 per cent to over 75 per cent; that is to say, the reflectivity for infra-red radiation falls from a value of 97 per cent to less than 25 per cent. For an opaque body, the sum of the reflectivity and emissivity at the same wave-length is 100 per cent. While a substantial change in emissivity has taken place with increases in thickness of the oxide coating up to about 30 x 10^{-5} in. the changes in reflectivity for visible light have been of minor character. Fig. 4 shows the effect on reflectivity for light, of increase in thickness of the oxide coating for two types of oxide-coated surfaces: one of these coatings is specular in character and the other diffuse, and the metal in each case is of different purity. However, even in the case of the coating showing the greatest change, the loss in reflectivity for a coating thickness of 30 x 10^{-5} in. was only from 86 to 75 per cent.

The reflectivity for visible light was measured by the method of A. H. Taylor,[14] which gives the total reflectivity directly without the use of any standard reflecting surface. The equipment employed was the instrument designed by Baumgartner,[15] which employs a light sensitive cell for measuring the brightness of the sphere wall.

[12]Methods of Testing Oxide Coatings on Aluminum, by J. D. Edwards. (*Proceedings, American Society for Testing Materials, 37,* Appendix II, Report of Committee B-7, 1937, p. 261.)

[13]Built-up Films of Barium Stearate and Their Optical Properties, by Katherine B. Blodgett and Irving Langmuir (*Phys. Rev., 51,* 1937, p. 964.)

[14]A Simple Portable Instrument for the Absolute Measurement of Reflection and Transmission Factors, by A. H. Taylor. (*Bureau of Standards Bulletin,* Bureau of Standards Scientific Paper No. 405, 1920.)

[15]A Light-sensitive Cell Reflectometer, by G. R. Baumgartner. (*General Elec. Rev., 40,* 1937, p. 525.)

Commercial aluminum foil with an emissivity of 3 per cent has a reflectivity of about 85 per cent for visible light.

The data of Fig. 3 show that while the thickness of a lacquer coating has an important effect upon the emissivity of the surface, the presence of a dye in the lacquer has no appreciable effect upon emissivity. The presence

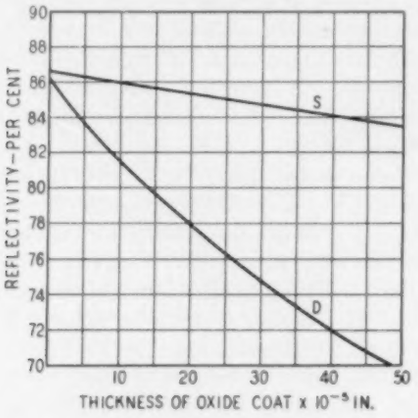


Fig. 4—Reflectivity for light of aluminum surfaces with electrolytically produced oxide films of different thickness; the *S* graph is for specular reflectors and the *D* graph is for diffuse reflectors

of the dye may, however, have a very profound effect upon reflectivity for light; data on this point are shown in Table 1. These figures show that a surface may have low reflectivity for light and fairly good reflectivity for infra-red radiation, or high reflectivity for light and poor reflectivity for infra-red radiation.

The lacquer films, measurements of which are shown in Table 1, are intensely colored. It is quite possible to coat with thinner lacquer films, and aluminum with these lacquer films may have high reflectivity (above 80 per cent) for infra-red radiation and still be decidedly colored because of the intense coloring action of the dye. Comparing the lacquers with black and white pigment, it will be noted that samples, showing as widely different reflectivities for light as 5 per cent and 67 per cent, show substantially the same reflectivity for infra-red radiation.

Effect of Surface Roughening

The assumption is commonly made that a highly polished metal surface is necessary for low emissivity. This assumption has, as a background, the classical theory which calls for an increase in emissivity as a surface is roughened.[16] Experimental measurements, however, demonstrate that roughening of a very pronounced degree may take place without any very substantial increase in the emissivity value. In Table 2 are given a series of emissivity values for foil which has been roughened in various ways. The samples include foil which had been embossed with various patterns for decorative effect, and also foil which had been roughened with an abrasive cloth (No. 120 Aloxite cloth). While it is true that the abrasive did not make the foil very rough, nevertheless its specular reflecting characteristics were completely destroyed. Among all of these samples,

[16]Heat Radiation of Technical Surfaces at Ordinary Temperatures, by E. Schmidt. (*Gesundheit.-Ing. Beihefte Series* 1, No. 20, 1927.)

Table 1—Comparison of Effect of Colored Lacquer on Aluminum Foil Upon Reflectivity for Visible and Infra-Red Radiation

Description of Samples	Thickness of Coating Inches x 10^{-5}	Reflectivity for Light Per Cent	Reflectivity for Infra-red Radiation[a] Per Cent
Lacquer, clear	20	80	57
Lacquer, clear	43	82	33
Lacquer, with black pigment	30	5	30
Lacquer, with black pigment	68	5	8
Lacquer, with white pigment	40	67	31
Lacquer, with white pigment	93	76	12
Lacquer, with yellow dye	20	76	43
Lacquer, with yellow dye	50	68	21
Lacquer, with blue dye	30	8	45
Lacquer, with blue dye	64	5	22
Lacquer, with red dye	27	28	39
Lacquer, with red dye	63	23	26

[a]Radiating surface at 100 F.

Table 2—Effect of Surface Roughening Upon Emissivity

Description of Sample	Emissivity at 100 F Per Cent
Embossed foil—Pattern No. 1	4.5
Embossed foil—Pattern No. 2	5.6
Embossed foil—Pattern No. 3	4.7
Embossed foil—Pattern No. 4	5.6
Embossed foil—Pattern No. 5	6.1
Bright foil roughened with abrasive cloth	4.4
Bright foil roughened with abrasive cloth	6.3
Bright foil roughened with abrasive cloth	6.6
Bright foil etched in hot sodium hydroxide solution for times of 0.5 to 2 min and dipped in nitric acid	3.5
"	4.0
"	5.0

the highest emissivity measured was 6.4 per cent. Tests were also made on foil which had been roughened by etching in sodium hydroxide solution for different times and finish-dipped in nitric acid, according to a procedure commonly employed in the aluminum industry. Six samples roughened in this way all had diffusing surfaces, and only one sample showed an emissivity as high as 5 per cent. All of these samples were so roughened as to destroy the polish and specular reflecting characteristics. Of course extreme methods of roughening might be employed which would give substantial increases in emissivity, but surfaces of that character are not ordinarily found.

Effect of Exposure Upon Emissivity

The very penetrating character of infra-red radiation was illustrated in the preceding section by the measurements on emissivity and reflectivity of oxidized aluminum surfaces. In quite another field, the penetrating characteristics of infra-red radiation are illustrated by the aerial photographs which have been taken by Captain Albert Stevens. With infra-red sensitized photographic plates, views of mountains as far as 331 miles distant from the photographer have been taken, though the mountain itself was invisible to the eye because of the intervening haze and fog.

Various surface films may be formed upon aluminum as the result of exposure to corrosive attack. Fortunately, the oxide film formed on aluminum in the atmosphere is quite impervious, and this limits its growth and thickness. This oxidation may therefore occur without an appreciable effect upon the reflectivity of the surface for infra-red radiation. The eye, of course, is no judge as to the infra-red radiation characteristics of a surface

Table 3—Emissivity of Aluminum Foil With Dulled Surfaces

Sample No.	Description of Sample	Emissivity Per Cent Radiating Surface at 100 F
1	Aluminum foil after 2 years' exposure to salt spray and moisture in boat-house on the seashore	10
2	Aluminum foil-covered cardboard taken from foil-insulated dry-ice cabinet; exposed to weather on beach at Coney Island from September, 1934, to April, 1935	4
3	Aluminum foil, chemically oxidized by treating with hot solution of sodium carbonate and chromate	9
4	Aluminum foil with thin lacquer coat heated to partially decompose lacquer and color it brown	10
5	Aluminum foil suspended vertically in laboratory for 3 years and measured with accumulated dust and fume	5

and surfaces, which to the eye may be dull and poorly reflecting, may still be good reflectors for radiation of long wave length.

As a quantitative measure of the effective difference between the reflection factor of aluminum for visible light and infra-red radiation, the following example may be cited: A sample of aluminum foil was exposed in the laboratory near a window for three years. This sample was allowed to accumulate dust and fume for that period, and its reflectivity for visible light was then measured without any preliminary cleaning. It was found after the three-year period of exposure to have lost 15 per cent in reflectivity for visible light; its infra-red reflection factor, however, had not changed appreciably. Originally, it was about 97 per cent, and with the three years' accumulation of dust and fume, its reflectivity had only been reduced to 95 per cent.

In June, 1938, Prof. Gordon B. Wilkes of Massachusetts Institute of Technology removed samples of aluminum foil insulation from the underside of a roof of a home in Wellesley Hills, Mass., where the insulation had been installed in November, 1933. Samples from each of the four layers of crumpled foil appeared to be in perfect condition and untarnished. The average emissivity of the four samples of foil which had been in service for five years was 5.4 per cent as compared with a value of 4.5 per cent which Professor Wilkes used for the new foil.

A series of emissivity measurements of aluminum, which had been exposed to the weather or chemically oxidized and treated in various ways, is given in Table 3.

Summary

1. A simple instrument for measuring emissivity is described and a variety of measurements presented to show the effect of surface films on the emissivity of aluminum at 100 F.

2. The oxide coating naturally present on aluminum is too thin to have an appreciable effect upon emissivity. Thick oxide coatings can be produced electrolytically, however, which substantially increase the emissivity. The relation between oxide thickness and emissivity for one type of coating is shown by a graph.

3. Thin lacquer films can be applied to aluminum, which are protective in nature but not thick enough to raise the emissivity above a value of about 10 per cent. The relation between thickness and emissivity is presented in a graph.

4. Colored dyes in lacquer films have little effect upon emissivity, but pigments have an appreciable effect.

5. The eye may be a good judge in appraising the reflectivity of a surface for visible light but cannot evaluate the infra-red radiation characteristics. A piece of aluminum may have high reflectivity for visible light and low reflectivity for infra-red radiation, or it may have only fair reflectivity for light and be an excellent reflector of infra-red radiation, depending on the presence or absence of surface films. It may also be a good reflector for both kinds of radiation.

6. Surface roughening does not necessarily produce a surface of high emissivity and a bright polished aluminum surface is not an inherent requisite for low emissivity.

7. Because the natural oxide film on aluminum is quite impervious and very slow to grow in thickness, substantial weathering of aluminum foil can occur with little increase in emissivity.

8. Samples of aluminum foil used for thermal insulation under various conditions of service have shown very little change in emissivity over long periods.

Fourth Edition of Refrigerating Data Book

The fourth edition of the *Refrigerating Data Book* has been published by *The American Society of Refrigerating Engineers*. The format and arrangement of the book follows its predecessors in this biennial series and was produced under the direction of the following committee: A. B. Stickney, *chairman;* J. C. Bergdoll, Arthur W. Ewell, George E. Hulse, V. L. Maleev, L. E. Philipp, C. D. Mackey, Charles H. Roe, Gardner Poole, F. O. Urban and George E. Wallis.

The fourth edition comes in 50 chapters, divided into eight parts, which can be referred to readily by thumb tabs. The parts are as follows: (1) Refrigeration (Principles); (2) Refrigerants; (3) Heat Flow and Insulation; (4) Air Conditioning; (5) Foods; (6) Domestic-Commercial Machinery; (7) Industrial Machinery; (8) Controls and Power.

The price of the book is $4.00 in the United States. Copies go to all members of the *ASRE* in all grades, included in their dues.

By special arrangement, ASHVE members can obtain this volume under the Exchange Service Plan recently renewed by the two societies.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the ASHVE Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, ASHVE Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.
F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

The Performance of Several Types of Bituminous Coal on Small Underfeed Stokers, by H. R. Limbacker and Ralph A. Sherman. Published by *Bituminous Coal Research Inc.,* 803 Southern Building, Washington, D. C. Mimeographed, 28 pages. Results of burning tests with eight coals in three domestic underfeed stokers, all of the screw-feed type.

•

Solid Smokeless Fuel, by M. Barash. *Gas World,* Vol. 109, pp. 129-136, Aug. 13, 1938. A review of methods of production, preparation and utilization of coke.

•

Heating at Norris, Tennessee, by Louis Grandgent, Charles F. Ellis and W. H. Purnell. Published by *Tennessee Valley Authority,* Department of Regional Planning Studies, Community Planning Division. Mimeographed, 42 pages.

•

Heating Costs for Temperatures Above 70 F. *American Artisan,* Vol. 107, No. 11, Nov. 1938, p. 31. Table on increased cost of heating a home when indoor temperature is above 70 deg. Table ranges from 68 F to 80 F. Cost ranges from minus 6.2 per cent to 31.0 per cent above heating cost at 70 F. Data taken from tests of oil burning equipment.

•

A Comparison of Costs of Heat Supply by Electrode Boiler or by Coke Boiler to the Same Group of Buildings, by S. Livingston Smith and C. H. Lander. *Journal of the Institution of Heating and Ventilating Engineers,* Vol. 6, No. 68, pp. 373-427, Oct. 1938. Report of heating cost tests conducted over a period of several years at the Imperial College of Science and Technology, England, where the same group of buildings was heated simultaneously with coke-fired boilers and with an electric thermal-storage system.

•

Proper Air Supply for Stokers, by S. H. Viall. *Coal Heat,* Vol. 34, No. 5, pp. 23-24, Nov. 1938. Directions for firing a stoker to obtain most efficient results from the coal.

•

The Permissible Working Time in High Temperatures and Humidities, by O. Ehrismann, and A. Hasse. *Arch. f. Gewerbepath u. Gewerbehyg.,* Vol. 8, pp. 611-638, 1938. Report of tests in a conditioned room at the Berlin Hygienic Institute on three men to determine the effect of temperature and air humidity on capacity and skin perspiration. Effective temperature better suited to determine capacity threshold than the Kata thermometer. Skin perspiration found to vary so widely that it was not predictable by wet or dry Kata or by effective temperature.

•

Physiological Effect of Prolonged Muscular Work, by E. Andrieu and G. Trechinos. *Travail Humain,* Vol. 6, pp. 11-28, 1938. Report of changes in respiration and circulation when hard work is carried on in the morning after sleep, and when the same work is done in the evening after a work day.

•

The Influence of Clothing on the Physiological Reactions of the Human Body to Varying Environmental Temperatures, by A. P. Gagge, C.-E. A. Winslow and L. P. Herrington. *The American Journal of Physiology,* Vol. 124, No. 1, Oct. 1938, pp. 30-50. Bibliography.

•

The Relative Influence of Radiation and Convection Upon the Temperature of the Clothed Body, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. Ibid, Vol. 124, No. 1, Oct. 1938, pp. 51-61. Bibliography. Report of tests on human subjects at the John B. Pierce Laboratory of Hygiene, New Haven, Conn.

•

The Measurement of Air Conditions, by Dr. Carey P. McCord and William N. Witheridge. *Journal of the American Medical Association,* Vol. 111, No. 18, Oct. 29, 1938, pp. 1647-1654. Description of the more important instruments and charts devised to measure human comfort and an appraisal of the reliability of various measures of atmospheric conditions for human comfort.

•

Engineering Problems Associated with the Improvement of Temperature and Humidity Conditions of the Atmosphere in Mines at Great Depth, by J. H. Dobson and Prof. W. J. Walker. The *Institution of Mechanical Engineers, Proceedings,* Vol. 139, pp. 185-235, 1938. Discussion of the production of satisfactory conditions for underground workers, during industrial mining operations where the wet-bulb temperature of the air of the mine attains 90 F. Use of water, brine or air for cooling.

•

Determination of Lead in Air and Identification of Industrial Dusts. *Preventive Engineering Series, Bulletin No.* 2, parts 6 and 7, the Air Hygiene Foundation of America Inc., Pittsburgh, Pa.

•

Base-metal Thermocouples; Their Characteristics, by M. R. Mandlekar and H. N. Banerjea. *Journal of the Society of Chemical Industry,* Vol. 57, No. 8, Aug. 1938, p. 276. Discussion of the characteristics and the permissible range of base-metal thermocouples.

•

The Ventilation of Flueless Rooms. *Journal of the Institution of Heating and Ventilating Engineers,* Vol. 6, No. 68, Oct. 1938, pp. 428-434. Report of ventilation tests in two rooms in the experimental house at the Building Research Station, England. The ventilation of a room was determined by measuring the increase of carbon dioxide after burning three standard candles in the room for three hours. Ventilation data are expressed as air changes per hour for flueless room, and room with flue and the same room with a ventilator.

•

Fan Laws and Outlaws, R. W. Smith and T. C. Lloyd. *Heating and Ventilating,* Vol. 35, No. 11, Nov. 1938, pp. 32-34. Discussion of fundamental relationships governing horse power required to move air. Fan dimensions in relation to ratings.

•

Infiltration Loss, Its Effect on Furnace-blower Capacity, by G. C. Ziliotto. *American Artisan,* Vol. 107, No. 11, Nov. 1938, pp. 52-53. Discussion shows that entire infiltration loss must be added to other heat loss for a building in estimating furnace capacity.

•

A Modern Watertight Steel House. *The Canadian Engineer,* Vol. 75, No. 19, Nov. 8, 1938, pp. 4-5. Description of a 41-ton sea-going residence believed to be the first all steel house ever constructed. The steel house was towed across the Illinois River on its own bottom. It contains five rooms and a garage. It was fully equipped with furnace, stoker, cooling system, laundry, plumbing and was partially furnished.

Pittsburgh—January 23-26

A FINE program has been provided for those who attend the Society's 45th Annual Meeting in the William Penn Hotel, Pittsburgh, Pa., January 23 to 26, 1939. Members of the Pittsburgh Chapter will act as hosts on this occasion. Registration headquarters will be established on the 17th floor of the William Penn Hotel and all members, guests and ladies are invited to obtain their credentials promptly after arrival.

After a meeting of Chapter Delegates on Monday morning there will be a Welcome Luncheon and immediately following this the first business session will be held in the Urban Room of the hotel.

The ladies will be entertained at tea on Monday afternoon and following the technical session there will be a Social Hour and in the evening a Get-together Dinner will be held in The Chatterbox. On Tuesday afternoon the ladies will have a Bridge-Luncheon at one o'clock and in the evening there will be informal entertainment after dinner and then the crowd will go to the Bavarian Grill.

On Wednesday evening the Annual Banquet and Dance is to be held in the ballroom of the William Penn Hotel and at that time the presentation of the F. Paul Anderson Medal will be made to Prof. Frank B. Rowley, Director of Engineering Experiment Station, University of Minnesota, and Pres. E. Holt Gurney will receive the Past Presidents Emblem.

On Wednesday morning there will be a trip to East Liberty for the ladies followed by a luncheon in the Schenley district and those who desire to may inspect the Cathedral of Learning, Foster Memorial and the Heinz Chapel.

The technical program will emphasize the research work of the Society, both at the Laboratory in the U. S. Bureau of Mines and in cooperating institutions. This meeting marks the 20th Anniversary year of the establishment of the Society's Research Laboratory and the program will fittingly commemorate this event.

The following men who have served as Chairmen of the Committee on Research have been invited to attend the Pittsburgh Meeting: George W. Barr, Villa Nova, Pa.; Perry West, Lexington, Ky.; J. Irvine Lyle, Syracuse, N. Y.; F. R. Still, New York, N. Y.; Thornton Lewis, New York, N. Y.; E. E. McNair, St. Petersburg, Fla.; W. H. Driscoll, Syracuse, N. Y.; H. P. Gant, Glenmoore, Pa.; S. R. Lewis, Chicago, Ill.; L. A. Harding, Buffalo, N. Y.; F. B. Rowley, Minneapolis, Minn.; C. V. Haynes, Philadelphia, Pa.; G. L. Larson, Madison, Wis.; John Howatt, Chicago, Ill.; A. P. Kratz, Urbana, Ill.; W. A. Danielson, Louisville, Ky.

Each of the Society's 25 Local Chapters has designated an

[Concluded on page 68]

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desks, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc. and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

Program

45th Annual Meeting

William Penn Hotel **January 23 to 26, 1939**

Sunday, January 22

10:00 a.m. Meeting of the Committee on Research.

2:00 p.m. Informal Reception of Visiting Members and Ladies, Pittsburgh Chapter Room (17th floor, William Penn Hotel.

2:00 p.m. Meeting of the Council.

Monday, January 23

8:30 a.m. Registration—Pittsburgh Chapter Room (17th floor).

9:30 a.m. Meeting of Chapter Delegates (Cardinal Room, 17th floor).

12:00 noon Welcome Luncheon—Members, Guests and Ladies.
Greetings by Mayor Scully of Pittsburgh.
Address by Charles E. Mayette.

2:00 p.m. Afternoon Session (Urban Room 17th floor)
Call to order by Pres. E. Holt Gurney.
Announcements.
Reports of Officers.
Reports of Council Committees.
Report of Committee on Research, W. L. Fleisher.
Introduction of Past Chairmen of Committee on Research.

Technical Papers—

Frictional Resistance to the Flow of Air in Straight Ducts by F. C. Houghten, J. B. Schmieler, John Zalovcik and Nicholas Ivanovic.

Air Conditioning in Industry by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber.

Report of Tellers of Election.

3:00 p.m. Ladies Tea (Forum Room, 17th floor).

5:00 p.m Social Hour (Pittsburgh Chapter Room, 17th floor).

7:00 p.m. Get-together Dinner, The Chatterbox (William Penn Hotel).

Tuesday, January 24

9:30 a.m. Morning Session (Grand Ballroom, 17th floor).

Physiological Sensations—

Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice by C.-E. A. Winslow.

Cardiac Output, Peripheral Blood Flow, and Blood Volume Changes in Normal Individuals Subjected to Varying Environmental Temperatures by F. K. Hick, R. W. Keeton, Nathaniel Glickman and H. C. Wall.

The Role of the Extremities in the Dissipation of Heat from the Body in Various Atmospheric and Physiological Conditions by Charles Sheard, Marvin M. D. Williams, Grace M. Roth and Bayard T. Horton.

Skin Temperatures of the Extremities and Effective Temperature, by Charles Sheard, Marvin M. D. Williams and Bayard T. Horton.

Air Conditioning Requirements of an Operating Room and Recovery Ward by F. C. Houghten and W. Leigh Cook, Jr.

1:00 p.m. Ladies' Bridge Luncheon (Cardinal Room, 17th floor).

2:00 p.m. Afternoon Session (Grand Ball Room, 17th floor).

Heat Transfer—

Some Reflection and Radiation Characteristics of Aluminum, by C. S. Taylor and J. D. Edwards.

A Study of the Heat Requirements of a Single-Glazed Test House and a Double-Glazed Test House by M. L. Carr, R. A. Miller, Leighton Orr and David Shore.

Heat Transfer in Storage Water Heaters by D. W. Nelson and A. A. Rosenberg.

Condensation of Moisture and Its Relation to Building Construction and Operation by F. B. Rowley, A. B. Algren and C. E. Lund.

5:00 p.m. Social Hour (Pittsburgh Chapter Room, 17th floor).

6:30 p.m. Past Presidents' Dinner.

9:30 p.m. Bavarian Singers and Dancers, William Penn Grill, Special Supper for A.S.H.V.E. Members.

Wednesday, January 25

9:30 a.m. Morning Session.

Fuels—

Responsibility of A.S.H.V.E. Towards Solid Fuel Industry by W. A. Danielson.

Selection of Solid Fuels from the Viewpoint of the Small Consumer by P. Nicholls.

Small Stokers by P. A. Mulcey and R. A. Sherman.

Performance of Stoker-Fired and Hand-Fired Warm-Air Furnaces in the Research Residence by A. P. Kratz, S. Konzo and R. B. Engdahl.

12:00 noon Luncheon Meeting Nominating Committee.

2:00 p.m. Afternoon Session (Urban Room, 17th floor).

Air Pollution—

Smoke Abatement—Where to Draw the Line by H. B. Meller.

Smoke Producing Tendencies in Coals of Various Ranks by H. J. Rose and F. P. Lasseter.

Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density by F. B. Rowley and R. C. Jordan.

7:00 p.m. Annual Banquet and Dance (Grand Ball Room, 17th floor). Presentation of F. Paul Anderson Medal. Presentation of Past Presidents Emblem.

Thursday, January 26

9:30 a.m. Morning Session (Urban Room, 17th floor).
Installation of New Officers.

Design Factors—

The Effect of Heat Storage and the Variation in Outdoor Temperature and Solar Intensity on the Heat Transfer Through Walls by J. S. Alford, J. E. Ryan and F. O. Urban.

Analysis of Weather Design Data in the United States by J. C. Albright.

Application of Summer Weather Data in Design by John Everetts, Jr.

The Flow of Heat Through Walls by F. E. Giesecke.

J. F. S. Collins, Jr.

Arthur McGonagle

C. M. Humphreys

R. A. Miller

T. F. Rockwell

F. C. McIntosh

R. J. J. Tennant

Mrs. J. F. S. Collins, Jr.

P. A. Edwards

R. B. Stanger

E. C. Smyers

M. L. Carr

Reactions of Office Workers to Air Conditioning in South Texas by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses.
Unfinished Business.
New Business.
Resolutions.
Adjournment.

12:30 p.m. Council Meeting.

Committee on Arrangements

JOHN F. S. COLLINS, JR., *General Chairman.*
ARTHUR MCGONAGLE, *Honorary Chairman.*
C. M. HUMPHREYS, *1st Vice Chairman.*
FABIAN C. MCINTOSH, *2nd Vice Chairman.*
M. L. CARR, *Chairman* Finance Committee.
T. F. ROCKWELL, *Chairman* Publicity Committee.
D. W. LOUCKS, *Chairman* Transportation Committee.
R. J. J. TENNANT, *Chairman* Inspection Committee.
P. A. EDWARDS, *Chairman* Technical Sessions Committee.
A. F. NASS, *Chairman* Attendance Committee.
MRS. JOHN F. S. COLLINS, JR., *Chairman* Ladies Committee.
R. B. STANGER, *Chairman* Entertainment Committee.
E. C. SMYERS, *Chairman* Banquet Committee.
J. EARL FRAZIER, *Chairman* Reception Committee.

[Concluded from page 65]

official delegate to attend the meeting and a special program of topical discussions will feature the conference of these delegates on Monday, January 23.

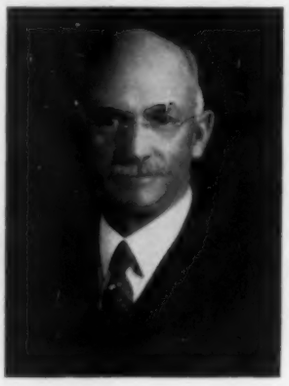
Plans are being made by many technical committees to hold meetings at Pittsburgh and reports will be presented at the various sessions.

Because of the central location of Pittsburgh and its convenience to a great majority of the membership, a big meeting is anticipated by the local group. Members have received Hotel Reservation Cards, information on the best motor routes to Pittsburgh and a hearty invitation from the Pittsburgh members to be on hand for the Society's 45th Annual Meeting.

Death of Warren Webster

The heating industry lost one of its pioneers in the death of Warren Webster who passed away on Wednesday December 21 in Georgetown University Hospital, Washington, D. C., at the age of 75.

Mr. Webster was President and General Manager of Warren Webster Co., Camden, N. J., for over fifty years. He was born June 25, 1863, in Philadelphia and attended the public schools of that city and prepared for the United States Naval Academy



at Annapolis. He entered business, however, after graduation from the Pierce Business College, and from 1887 was associated with his brothers in the heating business as Webster Brothers & Co., this firm being succeeded by Warren Webster & Co. in 1895. From the time of the organization of the company Mr. Webster served as President and the 50th Anniversary of the founding of the company was celebrated on June 23, 1938, at Camden, N. J., when Mr. Webster was the guest of honor at a testimonial dinner.

Mr. Webster was known throughout the heating industry for his pioneer work in developing the vacuum system of steam heating. He was the inventor and patentee of numerous devices for heating buildings and he purchased many other inventions and developments to perfect the Webster System. Until 1880 the common theory of steam circulation was that a constant pressure should be maintained throughout the entire system. In 1882 Mr. Webster acquired the Williams patent which introduced a new theory in the heating field and later Mr. Webster used the radiator trap based on the Hall Patent of 1891.

Mr. Webster established his first factory in Philadelphia and from a modest beginning in the manufacture of feed water heaters he expanded the business rapidly as vacuum heating systems came into vogue. From the proceeds of his sale of patents for a steam economizer and vacuum principle to a Belgian firm he erected a plant at Camden, N. J. The name Webster is known throughout the world in connection with heating and many large and well known buildings have this equipment including the original Palmer House in Chicago and its successor.

Mr. Webster was a life member of the ASHVE and he was originally elected to membership in 1899. He also held membership in the Manufacturers Club and the Penn Athletic Club of Philadelphia and the Island Heights, N. J., and the St. Petersburg Florida Yacht Clubs. Recently he retired from the Board of Trustees of Cooper Hospital, Camden, N. J., a position that he had held for many years.

Funeral services for Mr. Webster were conducted in Camden December 23. The Officers and Council of the Society extend their sincerest sympathy to the members of his family who survive, his widow, the former Frances M. Siegrist of Philadelphia; a son, Warren Jr., of Haddonfield, N. J.; two daughters, Mrs. W. Maynard Brown, of Haddonfield, and Mrs. J. Spencer Lucas, of Philadelphia; 11 grandchildren; a brother, A. Spencer Webster of Florida and Ocean Gate, N. J., and a sister, Mrs. George H. Musgrave of Arlington, Mass.

Cooling Tower Studies in California

A cooperative research project sponsored by the Society at the University of California, Berkeley, and directed by Dr. B. M. Woods, Chairman of the Technical Advisory Committee on Cooling Towers, Evaporative Condensers and Spray Ponds, has undertaken a program consisting of tower tests, general inves-

Cooling Tower at University of California

tigation of the status of evaporative condensers, field tests on water cooled roofs, and the analysis of data that will result in reliable and useful design methods.

As many problems in the design of cooling towers have only been partially examined in the past, it is the hope of the committee that the present investigation will produce design principles that will be effective in reducing future operating costs of cooling equipment. The behavior of a cooling tower depends upon many factors which include distribution of water at the tower entrance, slope of the elements of the decks, proper location of deck elements, and impurities and plant growths on the deck.

In the experimental work being carried on for the cooling towers, two of forced draft and two of atmospheric type are being used. The accompanying illustration shows part of the installation at the University of California.

34th Annual Meeting of ASRE

The *American Society of Refrigerating Engineers* held its 34th Annual Meeting at the Hotel Commodore, New York, December 6-8.

The newly elected Officers and Directors are as follows: Gardner Poole, Vice President, General Foods, Inc.—*President;* George Hulse, Chief Engineer, Safety Car Heating and Lighting Company—*Vice President;* Chester Lichtenberg, Engineer, General Electric Company—*Vice President;* William R. Hainsworth, Vice President, Servel, Inc.—*Treasurer;* Frank M. Cockrell, Publisher, *Air Conditioning & Refrigeration News*—*Director;* Charles S. Logan, District Representative, Superior Valve & Fittings Company—*Director;* B. E. Seamon, President, Westland Engineering Supply Company—*Director;* F. C. Stewart, Professor of Mechanical Engineering, The Pennsylvania State College—*Director;* Frank Zumbro, Engineer, Frick Company—*Director;* W. R. Woolrich, Dean of the College of Engineering, University of Texas—*Director.*

At the technical meetings the main subjects of discussion were air conditioning, refrigeration in industry, foods and commercial refrigeration, quick freezing and market research.

New Boston Office

The Boston office of the Air Conditioning Utilities Co. is now located at 436 Statler Building. The New York and Boston activities are under the direction of Harry W. Fiedler, factory representative for Julien P. Friez & Sons Division of Bendix Aviation Corporation, Baltimore, Maryland.

Chicago Symposium on Air Conditioning

The subject of comfort air conditioning attracted a crowd of over 1200 at the Engineering Building, Chicago, November 22nd, when a meeting was sponsored by the *Western Society of Engineers,* the Illinois Chapter of the ASHVE, the *Chicago Association of Consulting Engineers,* and the Chicago Section of the *American Institute of Architects,* and the *Air Conditioning Manufacturers Association.*

Nationally known leaders gave short talks covering a variety of phases of present day practice in comfort air conditioning. The Committee in charge of the program was: William J. Smith, Chairman; R. E. Hattis, Vice-Chairman; J. R. Vernon, John Aeberly, Leigh S. Keith, E. T. Murphy, R. L. Jones, and Leo J. Weissenborn.

The Importance of Correct Air Conditioning Standards was the subject of Dr. Irving S. Cutter, Dean of the College of Medicine, Northwestern University. Charles S. Leopold, Consulting Engineer of Philadelphia, spoke on Engineering Aspects of Air Conditioning. W. H. Carrier, Chairman of the Board, Carrier Corporation, Syracuse, N. Y., talked about the Development of Air Conditioning. Albert Buenger, Delco-Frigidaire Conditioning Corporation, Dayton, discussed Structural Items in the Architectural Design of Buildings. J. R. Hertzler, York Ice Machinery Corporation, York, Pa., spoke on Refrigeration as Applied to Air Conditioning. Elliott Harrington, General Electric Company, Bloomfield, N. J., gave Some Practical Applications of Air Conditioning, and William B. Henderson, Executive Vice President, *Air Conditioning Manufacturers Association,* Washington, gave a general review of Manufactured Products for Air Conditioning.

Second Oregon State Air Conditioning Conference

Announcement has been made that the second Oregon State Air Conditioning Conference will be held at Corvallis, March 23-25. Plans for the conference were announced by E. C. Willey, Assistant Professor of Mechanical Engineering of Oregon State College, after a meeting of the Committee, consisting of B. S. Farnes, J. Donald Kroeker, A. E. Finley, T. E. Taylor, Portland, and J. C. Othus, Corvallis.

C. P. Yaglou Addresses Massachusetts Chapter

December 20, 1938. The December meeting of Massachusetts Chapter was held at Massachusetts Institute of Technology, with 40 members present. Pres. James Holt presided and read a cordial invitation and telegram from the Pittsburgh Chapter inviting members to attend the Annual Meeting of the Society, January 23-26.

Prof. C. P. Yaglou, Harvard School of Public Health, spoke on Effects of Air Conditioning on Comfort, Health and Disease. This talk proved to be of particular interest to those present, according to the report of Secy. H. C. Moore.

Pittsburgh Chapter Elects Officers

December 7, 1938. Because of the approaching Annual Meeting of the Society to be held in Pittsburgh, January 23-26, the Pittsburgh Chapter meeting was a social gathering devoted to dinner and cards for members and their wives. Dinner was served in the Norse Room of the Fort Pitt Hotel at 7:00 p.m.

The only business transacted was the annual election of officers and immediately following dinner, Pres. J. F. Collins, Jr. called the meeting to order for this purpose. Secy. T. F. Rockwell announced the following candidates for office:

President—R. A. Miller
Vice-President—F. C. McIntosh
Secretary—T. F. Rockwell
Treasurer—L. S. Maehling
Board of Governors—J. F. Collins, Jr., *Chairman;* E. C. Smyers and E. H. Riesmeyer, Jr.

It was moved, seconded and passed that Secretary Rockwell cast a unanimous ballot for the officers nominated.

The retirement of Mr. Collins and the induction of President Miller were handled with due ceremony by a committee consisting of H. A. Canon, G. G. Waters, J. E. McLean and R. J. J. Tennant.

The remainder of the evening was enjoyably spent playing cards and dancing.

November 14, 1938. Pres. J. F. Collins, Jr., called the meeting of Pittsburgh Chapter to order at 7:50 p. m. in the private dining room of Stouffer's restaurant. Forty-eight members and guests were present. The minutes were read and approved as corrected and Prof. C. M. Humphreys acted in the place of Secy. T. F. Rockwell, who was absent.

Because of the plans of the Social Committee for a dinner dance in December, annual reports were presented at this meeting by Pres. J. F. Collins, Jr., Treas. L. S. Maehling, for the Research Committee by F. C. McIntosh, Degree Day Committee by F. C. Houghten, and Air Conditioning Code by F. B. Mahon.

Regular monthly reports were given by R. J. J. Tennant for the Social Committee and R. A. Miller for the Annual Meeting Committee.

The Nominating Committee submitted the names of the following candidates for election in December:

For President—R. A. Miller
For Vice-president—F. C. McIntosh
For Secretary—T. F. Rockwell
For Treasurer—L. S. Maehling

F. C. Houghten, director of the ASHVE Research Laboratory, was introduced by President Collins as the principal speaker on the subject of Air Conditioning in Industry. The earlier work of the Laboratory was quickly reviewed, and then Mr. Houghten described the current program of cooperative investigation. During these studies the reactions of employees in several air conditioned buildings to different combinations of temperature and relative humidity during recent cooling seasons were carefully determined. A close check of similar reactions under carefully controlled conditions was also made at the Laboratory. As this work is not complete, definite conclusions could not be presented.

The meeting adjourned at 9:55 p. m.

Montreal Chapter Meets With Engineering Institute; Discusses Health and Air Conditioning and Sound

December 12, 1938. Montreal Chapter held a meeting at the M.A.A.A. with an attendance of 42 members and guests. After dinner the meeting was called to order at 7:45 p.m. by Pres. F. J. Friedman, who extended a welcome to the guests.

The speaker of the evening was E. L. Anderson, director of research, American Blower Corp., Detroit, Mich., who was introduced by G. L. Wiggs. The subject of Mr. Anderson's address was Sound and Its Relation to Heating and Ventilating Apparatus. The lecture was illustrated with slides and after the address an open discussion was held.

E. T. Sampson moved a vote of thanks to the guest speaker on behalf of the Chapter.

According to Secy. C. W. Johnson the meeting was adjourned at 9:15 p. m.

November 21, 1938. A dinner and meeting of Montreal Chapter was held at the Windsor Hotel with an attendance of 32. Pres. F. J. Friedman called the assembly to order and extended a welcome to the guests who were present.

The speaker of the evening, Dr. F. G. Pedley, assistant professor of public health, McGill University, was introduced by President Friedman. Dr. Pedley discussed the subject of Health and Air Conditioning and stressed the fact that according to present day knowledge there is no conclusive evidence that air conditioning as known today is conducive to health, with the exception that it has been found to be definitely desirable for hay fever sufferers and also has high value in operating rooms and in hospital rooms for premature infants. Dr. Pedley stated that from the standpoint of bodily comfort and efficiency of factory workers air conditioning is definitely necessary and holds great promise, and that its value for improving efficiency and health should be carefully investigated by the engineering and medical professions.

After the address an open discussion was held and W. U. Hughes moved a vote of thanks to the speaker on behalf of the Chapter.

Minutes of the previous meeting were read and approved and Treas. F. G. Phipps urged payment of Chapter dues.

The meeting was adjourned at 9:30 p. m., according to the report of Secy. C. W. Johnson.

October 13, 1938. The opening meeting of the 1938-39 season of Montreal Chapter was held jointly with the Engineering Institute of Canada with a dinner meeting at the Windsor Hotel and the general technical session at the Engineering Institute.

The speakers of the evening were J. H. Walker, Detroit Edison Co., Detroit, who discussed Air Conditioning in a Glass Brick Building, and J. F. McIntire, Detroit, 1st vice-president of the ASHVE, who spoke on The Architect and the Engineer.

At the opening of the joint meeting, a welcome to the Montreal Chapter of the ASHVE was extended on behalf of the Engineering Institute by J. Challies, president, and a response was made by E. Holt Gurney, Toronto, president of the ASHVE.

W. L. Fleisher, chairman of the Society's Committee on Research, gave a brief resumé of ASHVE research activities.

A vote of thanks to the speakers was proposed by Mr. Coombe of Engineering Institute.

The joint chairmen of the evening were B. R. Perry, representing the Institute, and G. L. Wiggs, representing the ASHVE.

The meeting was adjourned at 10:00 p. m., according to the report of Secy. C. W. Johnson.

Fire Prevention and Health Aspects of Air Conditioning Discussed by Michigan

December 12, 1938. The meeting of Michigan Chapter was held at Huyler's L'Aiglon with 50 members and guests attending the dinner and approximately 70 at the meeting.

Due to the illness of Pres. F. J. Linsenmeyer, Vice-Pres. W. C. Randall presided and called the meeting to order at 7:30 p.m. Roll call reports of the secretary and chairman of the Membership Committee were in order.

The first speaker of the evening was R. L. Loughead, chief inspector, Michigan Inspection Bureau. He described the necessary features an air conditioning system should have in order to be free from fire hazard. He also pointed out that recent fires which have occurred in ventilating systems could have been prevented if these provisions had been included.

Otto Lindemeyer, fire marshal, City of Detroit, discussed the provisions mentioned in Mr. Loughead's paper that were related to fire prevention.

The second speaker of the evening was G. F. Emery, chief inspector, Bureau of Building and Safety Engineering, City of Detroit, who discussed the Proposed Air Conditioning Code for the City of Detroit. Mr. Emery pointed out that an air conditioning code should include the following items: design, construction, installation, and maintenance.

Discussion of the subject was given by E. L. Hogan, S. S. Sanford and Mr. Strand.

The meeting was adjourned at 10:30 p.m. according to the report of Secy. G. H. Tuttle.

November 7, 1938. Approximately 100 members and guests were present at the regular meeting of Michigan Chapter held at Huyler's L'Aiglon, Detroit.

Pres. F. J. Linsenmeyer called the meeting to order and asked for roll call and the reports of the secretary, treasurer, and program chairman.

Secy. G. H. Tuttle read an amendment to Article VI of the Chapter's By-Laws relative to the Chapter's representative and alternate on the Society's Nominating Committee. It was moved, seconded and passed unanimously that the amendment be adopted.

A. V. Hutchinson, secretary of the ASHVE, was present and spoke briefly on the Annual Meeting of the Society to be held in Pittsburgh, January 23-26.

Prof. C. P. Yaglou, Harvard School of Public Health, Boston, Mass., was the guest speaker and his subject was Health Aspects of Air Conditioning. Professor Yaglou pointed out the interesting fact that recent tests have proved that relative humidity has little effect upon human health and comfort for winter air conditioning as long as it is above 20 per cent.

Active discussion of the subject was given by Dr. Henry F. Vaughan, commissioner, Department of Health, City of Detroit, Mr. Witherage of the Herman Kiefer Hospital, and Dr. H. Lee Simpson of the *Wayne County Medical Society*.

The meeting was adjourned by President Linsenmeyer at 10:30 p. m. with everyone present giving a rising vote of thanks to Professor Yaglou.

Southern California Has Large Meeting

November 9, 1938. The Southern California Chapter held its second meeting at the Colonial Banquet Room in Los Angeles, with 156 members and guests in attendance.

After the regular business of the meeting was concluded, Pres. H. M. Hendrickson introduced F. J. Kurth, of Anemostat Corp. of America, who delivered a talk on Air Distribution. Mr. Kurth's talk contained information which was useful and educational to the members and guests. A visual demonstration of air flows and currents in a room added to its interest.

Canadian Engineer Discusses Hot Water Heating

November 10, 1938. The November meeting of the Philadelphia Chapter was held at the Engineers Club, and was preceded by a dinner attended by 77 members and guests.

The treasurer's report and the minutes of the previous meeting were read and approved.

Edwin Elliot reported on the activities of the membership committee and R. F. Hunger announced the program for the December meeting.

Pres. H. H. Erickson then introduced the speaker of the evening, G. L. Wiggs, consulting engineer, Montreal, Can., who presented a very interesting talk on Modern Hot Water Heating Practice. Mr. Wiggs' address was followed by a lively and constructive discussion.

The meeting adjourned at 10:00 p. m.

Air Conditioning in Industry Subject of Toronto Meeting

December 5, 1938. The regular meeting of the Ontario Chapter was held in the Royal York Hotel, Toronto, where dinner was served at 6:30 p. m. and 58 members and guests were present.

Pres. H. B. Jenney called upon E. R. Gauley, who spoke of the recent death of one of the Chapter's past presidents and one of its most popular and active members, W. P. Boddington.

The minutes of the preceding meeting were read by Secy. H. R. Roth and accepted as read.

President Jenney asked for nominations for Ontario Chapter's representative and alternate on the Society's Nominating Committee. A. G. Ritchie nominated H. R. Roth as representative and C. Tasker as alternate. The motion was seconded by Thomas McDonald and carried.

H. H. Angus was called upon by President Jenney to report for the Code Committee. Mr. Angus announced that the revised code on hot water and steam heating for residences had been submitted to the Board of Governors for their advice on the proper way of handling this code in the future. It was stated that the warm air heating code would be the next to come up for discussion.

The Society's Annual Meeting in Pittsburgh, January 23-26, was brought to the attention of the meeting by President Jenney.

J. W. O'Neill introduced the principal speaker of the evening, A. E. Stacey, Jr., New York, N. Y., who gave an illustrated talk on Air Conditioning in Industry. An interesting discussion followed Mr. Stacey's address. E. Holt Gurney, president of the ASHVE, expressed a vote of thanks on behalf of the Chapter to Mr. Stacey for his talk.

Insulation Topic at Massachusetts Meeting

November 15, 1938. The November meeting of the Massachusetts Chapter was held at the Massachusetts Institute of Technology and was preceded by a supper at Walker Memorial. The meeting was attended by 35 members.

The speaker of the evening was Prof. G. B. Wilkes, of M. I. T., who spoke on Some Problems in the Use of Insulation. This proved to be a very timely topic as indicated by the discussion after the talk. Besides various types of insulation, the effect of various positions with reflective insulation and the use of vapor barriers were discussed.

Some Hints from the "Trouble Shooters" Notebook Entertain Chicago Members

November 14, 1938. The November meeting of Illinois Chapter was held at the Merchants and Manufacturers Club in Chicago. Attendance at dinner was 104 members and guests with 120 attending the meeting, which was called to order by Pres. J. R. Vernon.

J. R. Borling, N. W. Swanson and C. A. Storch were appointed tellers to count the ballots for election of a member to fill the vacancy on the Board of Governors. Later Mr. Swanson announced the result of the ballot which was unanimous for W. A. Kuechenberg, who was therefore declared duly elected and he immediately assumed office.

Announcement was made by A. E. Christophersen of the dinner dance in honor of John Howatt at the Stevens Hotel on December 17.

President Vernon introduced W. J. Smith, chairman of the Program Committee, Chicago Chapter of AIA, who spoke of the Air Conditioning Symposium and called on R. E. Hattis, who read a detailed announcement of this event.

Dean Rogers of Lewis Institute and Mr. Turner, chairman of the Chicago Section of *ASME*, were introduced to the Chapter. Albert Buenger, Dayton, Ohio, chairman of the ASHVE Guide Publication Committee, was present and gave a brief resumé of the committee's work in compiling THE GUIDE 1939.

After the reading of minutes of the October meeting was dispensed with, Secy. M. W. Bishop read an announcement of an Air Conditioning Seminar at the University of Minnesota.

I. E. Brooke, chairman of the Membership Committee, introduced the following new members: A. J. Keating, C. A. Gustafson and J. M. Bradley.

V. L. Sherman, chairman of the Meetings and Publications Committee, announced the speakers for the November meeting and reported on the work of his committee.

President Vernon called on John Howatt to introduce the speaker, S. R. Lewis, consulting engineer, whose subject was

A Heating and Cooling Man Discusses His Troubles. Mr. Lewis drew on his wide experience in the design of successful heating and cooling installations, as he explained, not to extol their virtues but to tell about the things he would do differently were he to repeat the performance.

The lack of adequate zoning to compensate for variations in occupancy and heat reception from manufacturing processes, and failure to provide not only ample access doors, but also sufficient clearances to make the doors easily usable for maintenance, will often prove sources of complaint in an otherwise satisfactory installation.

The use of well water for pre-cooling and condensing purposes will result in large savings in both first and operating costs, but thought should be given to the possible presence of corrosive impurities such as hydrogen sulphide in such water supply, which may cause leaks in condenser tubes resulting in loss of the refrigerant and flooding of the system. If such impurities are present, the answer would seem to be to limit the use of this water to pre-coolers and to use evaporative condensers with city water make-up to accomplish condensing. The tonnage saving accomplished by the pre-cooler would more than offset the occasional replacing of pre-cooler coils, especially since no such damage would be caused by leaks at this point as would be caused by leaks in the condenser itself.

Mr. Lewis pointed out that in original design it is not enough to work toward the intended or existing conditions. Account must be taken of changes in use or occupancy of conditioned areas, as in the case of a department store in which the basement and first floor were supplied from a common central fan system. The heat gains indicated that air supplied at the same temperature would maintain both spaces at the proper conditions, and this was actually the case during the first summer of operation. However, before the second summer, the owner, finding the basement was comfortable and attractive, proceeded to double the illumination and to attract many more people. As a result, when the basement was then maintained at a comfortable temperature, the first floor was entirely too cool. The solution in this case was to install an entirely separate supply system for the basement, which could have been done at the start as easily as not.

Mr. Lewis brought out many other points including the necessity of adequate exhaust to keep down odors; fresh air intakes of sufficient size to permit the use of all outdoor air in order to take advantage of the many days when outdoor conditions are nearly perfect; use of reheater coils for small private offices supplied from central systems; conversion of antiquated or improperly maintained vapor heating systems to forced hot water; construction of tighter or leak-proof fresh air dampers; and elimination of short abrupt duct connections on both inlet and discharge sides of fans.

After Mr. Lewis' talk, a lively discussion followed in which J. J. Aeberly, R. E. Hattis, W. A. Kuechenberg and others took part.

October 10, 1938. Illinois Chapter opened the 1938-39 season with a meeting at the Merchants and Manufacturers Club, Chicago. Eighty-one members and guests were present at dinner and several others came later to attend the meeting.

Pres. J. R. Vernon called the meeting to order and, after giving a brief welcome, proceeded to the business of the meeting.

Announcement was made of the Air Conditioning Symposium on November 22 and of the Society's Annual Meeting, January 23-26.

A motion was made and carried endorsing the action of the committee in selecting the Merchants and Manufacturers Club as the Chapter's meeting place.

I. E. Brooke, chairman of the Membership Committee, announced the names of three new members, Prof. M. K. Fahnestock, R. T. Miller and W. W. Voss.

A brief report of the activities of the Meetings and Publications Committee was made by V. L. Sherman.

Secy. M. W. Bishop read the resignation of J. H. Van Alsburg as a member of the Board of Governors, due to the fact that his business activities prevent him from being in the city except for brief periods. The resignation was accepted with regret and a motion was made and carried that the secretary write to Mr. Van Alsburg, expressing the Chapter's regrets.

In accordance with the Constitution, President Vernon called for nominations from the floor to fill the vacancy on the Board of Governors. E. P. Heckel nominated W. A. Kuechenberg, which was seconded by H. G. Chapin and J. J. Aeberly. Since there were no other nominations, Mr. Kuechenberg was named to appear on the ballot to be sent to members and counted at the next meeting.

John Howatt read the report of the Legislative Committee on the proposed Illinois Engineering Council. After some discussion it was moved and carried that the secretary write to the headquarters office for permission for the Chapter delegates to sit in as observers at the Illinois Engineering Council meeting, pending action of the National Council on Illinois Chapter membership in this body.

Mr. Howatt presented a life membership certificate to J. J. Finan, who responded with a brief talk on what the Society has meant to him.

President Vernon called on J. H. Milliken to introduce the speaker of the evening, J. R. Allen, chairman of the Research Committee on Industrial Hygiene for *American Foundrymen's Association.*

Mr. Allen's address was on the subject of Ventilating Research in Industrial Hygiene and he prefaced his remarks by making a comparison of the objectives and engineering of heating and ventilating engineers and industrial sanitation engineers. According to the speaker, ventilating engineers use a rectangular duct, whereas industrial sanitation engineers use round duct almost exclusively; the ventilating engineer is most interested in supply systems and uses low velocities, the industrial engineer uses almost 100 per cent exhaust systems with minimum velocities of 3000 fpm and frequently goes to 4500 to 6000 fpm. The ventilating engineer is interested in air cleaning devices but usually deals in light concentration as in atmospheric dust, while industrial men are concerned with heavy concentrations and frequently with corrosive and explosive particles; the heating and ventilating engineer frequently recirculates air, whereas the industrial exhaust system exhausts all the air to a point outside the building.

Mr. Allen spoke of the background of the *American Foundrymen's Association* and of the reasons for the formation of their research and code committee, which has to date prepared and published three codes as follows: Code of Recommended Practices for Grinding, Polishing and Buffing Equipment Sanitation; Code of Recommended Practices for Testing and Measuring Air Flow in Exhaust Systems; and Recommended Good Practice Code and Handbook on Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems. The speaker outlined these codes, pointing out many interesting facts and conclusions which the committee has incorporated in their codes.

At the conclusion of Mr. Allen's address, a general discussion was held in which J. J. Aeberly, J. F. Tobin, S. I. Rottmayer and R. E. Hattis took part, before adjournment at 10:20 p. m.

Henry Smith Downe Dies

Services were held at the American Cathedral in Paris for Henry Smith Downe, where he died on December 8, 1938, after a long illness.

Mr. Downe, a Life Member of the ASHVE, having joined the Society in 1895, was born at Mason, N. H., on June 1, 1861. After his graduation from high school in Fitchburg, Mass., he entered the Worcester Polytechnical Institute, from which he received his B.S. and M.E. degrees.

In his early life he was connected with the Simonds Rolling Machine Co. at Fitchburg, and other concerns engaged in the manufacture of cast-iron boilers. In 1888 he became vice-president of the National Hot Water Heater Co., and later he was salesman for the American Boiler Co., which was merged in

1895 with the American Radiator Co. At this time he established an office in London, and in addition to developing the heating business in England he succeeded in introducing it in the principal countries on the Continent. As the sale of heating products increased, companies were formed and factories built in England, France, Germany, Belgium, Italy, Austria, Switzerland and Spain. Mr. Downe then became the European Director for the American Radiator Co.

He was one of the most prominent members of the American colony in Paris, belonging to the important American clubs in both Paris and London. He was also a member of the *ASME*, and was decorated by the French Republic as an officer of the Legion of Honor. While in Paris he was active in establishing the American Hospital, and maintained great interest in its development until his death. He paid an annual visit to the United States and was well known in business and social circles.

Death of E. T. Keefe

The death of E. T. Keefe, president of Underground Steam Construction Co., Boston, Mass., occurred on December 1 at his home in Newton, Mass. He was a native of Somerville, Mass., and attended the Boston Latin School.

Mr. Keefe was well known to those engaged in the heating and ventilating industry, as for the past 15 years he had been president of his company, engineers and contractors, serving principally power plants, district steam heating companies, state institutions and colleges on underground steam distribution lines. Prior to that time he had carried on a similar engineering and contracting business under his own name.

In 1931 Mr. Keefe was elected a Member of the American Society of Heating and Ventilating Engineers and the Officers and Council members sincerely regret the loss of an active and loyal member.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 35 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by January 16, 1939, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Allensworth, J. E., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	P. A. Edwards Leighton Orr
Armstrong, E. T., Grad. Student, Mass. Inst. of Tech., Cambridge, Mass.	James Holt C. W. Berry (*A.S.R.E.*)	C. E. Fuller (*A.S.M.E.*) Earle Buckingham (*A.S.M.E.*)
Avery, Ledyard, Mgr., Air Cond. Dept., Schumacher-Mackenzie, Ltd., Winnipeg, Man., Can.	J. B. Steele D. F. Michie	A. MacKenzie (*Non-Member*) J. H. Schumacher (*A.I.E.E.*)
Bailey, F. A., Jr., Owner, Bailey's Refg. Sales & Service, Charleston, S. C.	C. L. Templin L. L. Barnes	C. E. Brockinton T. T. Tucker
Baxter, W. E., Vtg. & Air Cond., W. E. Baxter, Ltd., Montreal, Que., Can.	G. L. Wiggs L. C. A. Walford	W. U. Hughes C. W. Johnson
Bishop, J. W., Mgr., Air Cond. Div., Can. General Elec. Co., Ltd., Toronto, Ont., Can.	J. H. Daynes H. R. Roth	L. P. Baker J. H. Fox
Braniff, P. R., Secy. & Treas., Braniff Engrg. Co., Oklahoma City, Okla.	E. F. Dawson A. A. Hoppe	C. Z. Woodworth E. W. Gray
Buckley, M. L., Estimator & Supt., Phillips Getschow Co., Chicago, Ill.	John Howatt R. M. Getschow	J. C. McClellan M. W. Bishop
Clo, H. E., Sales Engr., Air Filter & Equip. Corp., Chicago, Ill.	O. J. Greenwood J. H. Milliken	M. W. Bishop I. E. Brooke
Freeman, A. W., Sheet Metal Worker, Max Freeman, Jackson Heights, L. I., N. Y.	G. A. Belsky T. W. Kunzog	E. R. Snavely B. A. Jacobi (*Non-Member*)
Funck, E. H., Sales Mgr., Johnson Fan & Blower Corp., Chicago, Ill.	J. R. Vernon J. J. Aeberly	M. S. Bamond J. C. Hines
Gable, H. R., Engr., Mayflower-Lewis Corp., Minneapolis, Minn.	R. L. King F. B. Rowley	A. B. Algren R. C. Jordan
Goff, J. A., Dean, Towne Scientific School, Univ. of Penna., Philadelphia, Penna.	A. C. Willard W. L. Fleisher	F. R. Bichowsky M. K. Fahnestock
Gregerson, George, Lab. Engr., Industrial Training Corp., Chicago, Ill.	E. V. Hill J. J. Aeberly	A. H. Zimmerman W. D. Fleak
Hubbuch, N. J., Jr., Student, Univ. of Dayton, Dayton, Ohio.	R. A. Gonzalez I. C. Baker	Adam Hofmann (*S.P.E.E.*) A. R. Weber (*S.P.E.E.*)
Johnson, C. E., Sales Engr., Fox Furnace Div., Amer. Rad. Co., Greensboro, N. C.	J. E. Maynard L. B. Miller	C. W. Nessell V. B. Kofoed

REFERENCES

KUCINSKI, W. V., Theory Instructor, Atlantic Utilities, Inc., Kearney, N. J.

KUNZOG, T. W., Air Cond. Engr., Northern Air Cond. Co., Jersey City, N. J.

LARUE, J. A. D., Mgr., Roofers, Inc., Montreal, Que., Can.

LIFSHITZ, HYMEN, Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

LINSKIE, G. A., Engr., Joe Hoppe, Inc., Dallas, Texas.

MACHEREL, FERDINAND, Consulting Engr., Algiers, French North Africa.

MARKLAND, C. E., Mech. Engr., University of Illinois, Urbana, Ill.

MARSHALL, S. C., Chief Engr., Mayflower-Lewis Corp, Minneapolis, Minn.

MERENS, S. H., Sales Engr. & Estimator, T. H. Litvin Plbg. & Htg. Co., Chicago, Ill.

PEACOCK, G. S., Htg. Engr., Univ. of Pittsburgh, Pittsburgh, Pa.

PETTY, C. E., Sales Engr., U. S. Radiator Corp., Charlotte, N. C.

PUGH, D. C., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

REED, F. J., Asst. Prof. of Mech. Engrg., Duke University, Durham, N. C.

ROSEBY, T. A., Chief Design Engr., Carrier Australia, Ltd., Sydney, N. S.W., Australia.

SAGINOR, S. V., Mech. Engr., Carbide & Carbon Chem. Corp., S. Charleston, W. Va.

SLUTSKY, D. J., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

SPITZLEY, J. H., Junior Member, R. L. Spitzley Htg. Co., Detroit, Mich.

STARK, C. E., Vice Pres., Montclair Plbg. Supply Co., Montclair, N. J.

SYMONDS, E. S., Mgr., Abain Engrg., Ltd., London, E. C. 2, England.

CANDIDATES

Proposers	*Seconders*
W. S. Baver (*A.S.C.E.*)	Chester Mueller (*Non-Member*)
J. L. Averill (*Non-Member*)	Herbert Simon (*Non-Member*)
A. A. Adler	W. E. Heibel
Arthur Ritter	E. R. Snavely
F. H. Madely	J. B. Dykes
C. W. Johnson	G. L. Ballantyne
C. M. Humphreys	P. A. Edwards
T. F. Rockwell	F. C. McIntosh
G. C. MacEachin	J. J. La Salvia
Albert Buenger	J. L. Cassell (*A.S.M.E.*)
I. P. Gramatges (*Non-Member*)	A. L. Debes (*Non-Member*)
Gaston Veagnoux (*Non-Member*)	J. N. Elwell (*Non-Member*)
A. P. Kratz	C. W. Nessell
M. K. Fahnestock	W. H. Severns
R. L. King	A. B. Algren
F. B. Rowley	R. C. Jordan
S. L. Haas	W. B. Graves
F. C. Black	M. W. Bishop
P. A. Edwards	T. F. Rockwell
J. F. S. Collins, Jr.	H. A. Beighel
R. B. Rice	I. F. McIntire
R. F. Connell	E. H. Brandt, Jr.
C. M. Humphreys	P. A. Edwards
T. F. Rockwell	Leighton Orr
T. C. Cooke	H. H. Hill
R. B. Rice	W. M. Wallace, II
E. G. Carrier	R. E. Cherne
R. V. Dunne	A. A. Berestneff (*A.S.R.E.*)
H. R. Sago (*A.S.M.E.*)	C. P. Monday (*W.V.S.P.E.*)
F. S. Twitt (*N.S.P.E.*)	Ray Thomas (*Non-Member*)
C. M. Humphreys	P. A. Edwards
T. F. Rockwell	R. B. Stanger
R. L. Spitzley	F. J. Feely
R. D. Randall	J. S. Kilner
C. R. Morris	Walter Johnson (*Non-Member*)
W. M. Hedden	R. C. Cook (*Non-Member*)
W. L. Fleisher	John Everetts, Jr.
A. E. Stacey, Jr.	A. V. Hutchinson

CANDIDATES ELECTED

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BERRIDGE, W. W., Sales Engr., McColl-Frontenac Oil Co., Ltd., Montreal, Que., Can.

BUSSE, HERBERT, Chief Engr., Fisher & Co., Detroit, Mich.

CRONEY, P. A., Chief of Mech. Section, U. S. Housing Authority, Washington, D. C.

CROPPER, R. O., Operating Engr., War Dept., Ft. Knox, Ky.

DICKASON, G. D., Pres. & Treas., Genesee Htg. Service, Inc., Rochester, N. Y.

RABE, A. E., Pres., Condition-Aire Corp., Albany, N. Y.

SCHWARTZ, N. E., Gen. Mgr., Sidles Co., Airtemp Div., Omaha, Nebr. (*Advancement*)

VAUGHAN, L. L., Prof. Mech. Engrg., North Carolina State College, Raleigh, N. C.

ASSOCIATES

FERGUSON, J. H., Engr. in Charge, John Ferguson Plbg. & Htg. Co., Cleveland, Ohio.

GRIFFITH, C. A., Htg. Contractor, Cumberland, Md.

PARKER, R. A., Secy.-Treas., Parker-Carpenter, Inc., San Francisco, Calif.

TER WEEME, ALBERT, Sales Engr., N. V. Radiatoren, Amsterdam, Holland.

THOMAS, B. A., Mgr. Htg. & Engrg. Dept., Crane Co., Tampa, Fla. (*Reinstatement*)

TRENNER, KELVIN, Mgr., Engrg. Dept., Staats Coal Co., Norristown, Pa.

JUNIORS

EZZ-EL-DIN, KAMAL, Htg., Vtg. and Air Cond., Office of W. W. Nobbs, London, W. 14, England.

SANDERS, C. M., JR., Air Cond. Syndicate Repr., Westinghouse Elec. & Mfg. Co., Chicago, Ill.

SELTZER, P. A., Engr., Bryant Air Cond. Corp., Philadelphia, Pa.

STOTZ, R. B., Sales Engr., Frigidaire Div., Atlanta, Ga.

STUDENTS

LESCH, R. T., Student, University of Minnesota, Minneapolis, Minn.

WEISSBLATT, NORMAN, Student, N. Y. Technical Inst., Brooklyn, N. Y.